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The JANUARY, 1932

Heating · Piping and Air Conditioning

IN THIS ISSUE

John Howatt's article on methods of promoting school plant operating economy—a valuable paper.

●

Ralph E. Phillips and Donald T. Robbins describe the air conditioning of the Los Angeles stock exchange building and discuss zoning, cost of installation, cost of power, etc.

●

Temple C. Patton tells how to select economical thickness of insulation for piping and flat surfaces. Charts simplify the problem. A helpful article.

●

E. H. deConingh writes on piping for pneumatic conveying; S. B. Redfield solves vacuum pump problems; W. G. Frank classifies and describes air-borne impurities in his article.

●

T. H. Rea shows how traps save fuel; shut-off valves is W. H. Wilson's subject in Practical Piping Problems.

Journal Section of
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Heating · Piping
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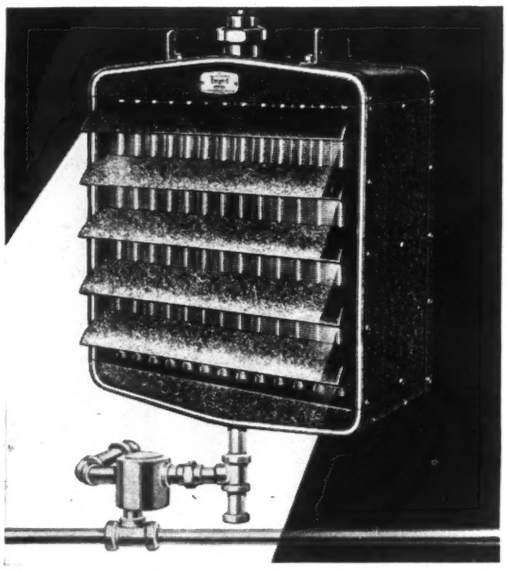

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The EDITOR'S PAGE

Heating · Piping and Air Conditioning

Temple C. Patton's article in this issue contains two charts which are of considerable aid in determining the economical thickness of insulation for piping and for flat surfaces. In his paper, Mr. Patton has discussed in detail three type cases: one insulation material applied to a flat surface; two insulation materials applied to a flat surface; and one insulation material applied to circular piping.

Examples illustrating each type case are presented in detail and show the application of these charts.

John Howatt's article in this issue points out ways to secure economy in the operation of school heating and ventilating plants. This paper was prepared as the result of experiences in supervising over 400 public school buildings. Careful records of fuel costs, money spent for maintenance, etc., have been kept for the past ten years, and these data are analyzed and discussed by Mr. Howatt.

This article is one of the series appearing regularly in HEATING, PIPING AND AIR CONDITIONING and stressing ways and means of designing, installing and maintaining heating, piping and air conditioning equipment to obtain economy. Each paper is prepared by an authority; last month, an article told how savings of 45 per cent in power to operate a central station-type air conditioning system were made by a method of intermittent control.

In coming issues, other specific subjects of peculiar interest to the heating, the piping and the air conditioning engineer will be treated.

When whatever committee that does that sort of thing was extolling the advantages of Chicago as the location for the next Republican convention, the point that the meeting would be held in the air conditioned Chicago Stadium was stressed by Edward N. Hurley. According to newspaper reports, Mr. Hurley played this trump card after the mayor of Atlantic City—which also wanted the convention—stated that he feared for the safety of the delegates in Chicago.

Anyhow, as you know, the G. O. P. will meet in Chicago next June. The moral must be that Republicans would rather take a slim chance of being shot at than to face attendance at a convention held without the benefits of air conditioning. Probably Democrats would, too.

A proposed feature of the annual meeting of the American Society of Heating and Ventilating Engineers this month is an inspection trip to Akron to see the home of the airship "Akron."

In this issue we publish several views of the equipment which helps to make the "Akron" the marvelous ship it is.

Variations in the nature of the occupancy in the different areas of the Los Angeles stock exchange building made the design of the air conditioning system for this building a problem which required careful consideration. The description of how this structure is supplied with conditioned air makes a story which is of no small interest to air conditioning engineers. It was written by Ralph E. Phillips, the consulting engineer, and Donald T. Robbins, chief engineer with Mr. Phillips. In addition to the discussion of the design and lay-out, data on cost of installation and cost of power are included.

Mr. Phillips is a firm believer in the "advertising value" of air conditioning; provisions for comfortable atmospheric conditions aid in securing tenants and are of great value in holding tenants. For this reason—and to aid efficient operation—machinery rooms are well-lighted and an endeavor is made to arrange the equipment logically and attractively.

We hereby express our thanks to the secretary of a company which operates several hotels, who wrote us recently as follows:

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Smiling modestly to ourselves, we hasten to publish his letter.

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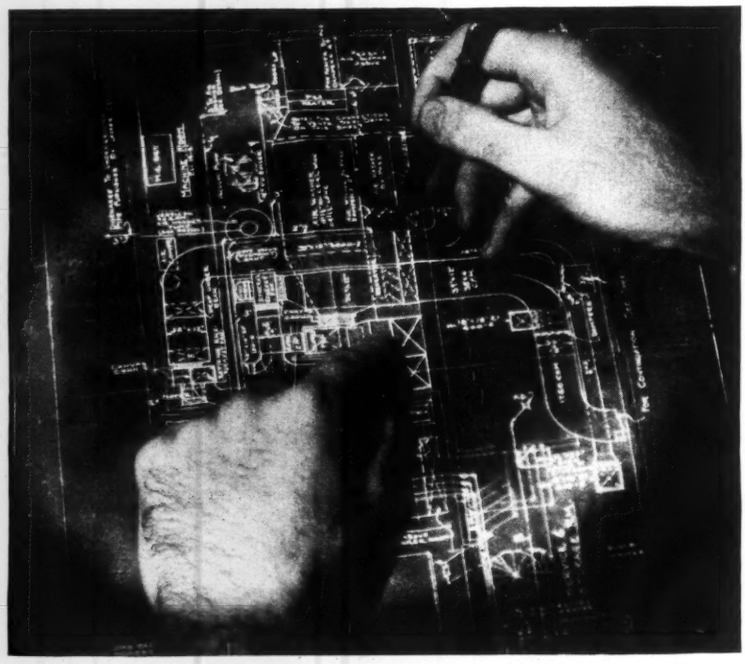

Control of by-pass dampers on a Split System installation. Left hand indicates location of duct thermostat beyond the fan. Right hand points to location of reversing damper controller on by-pass dampers.

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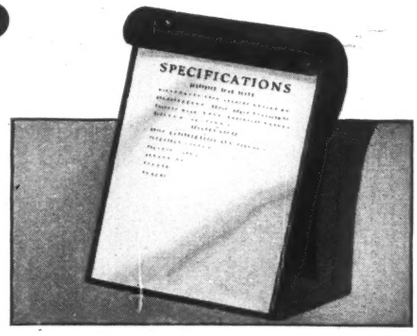

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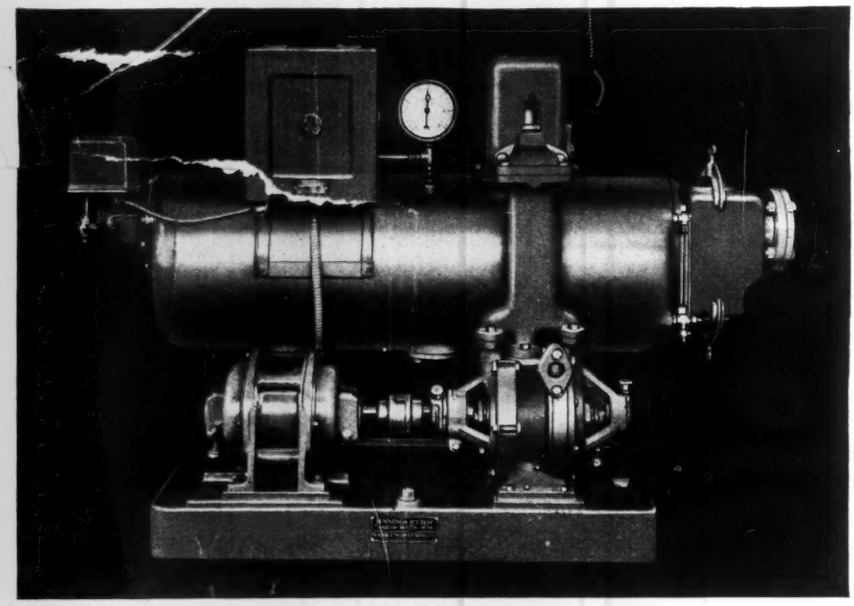

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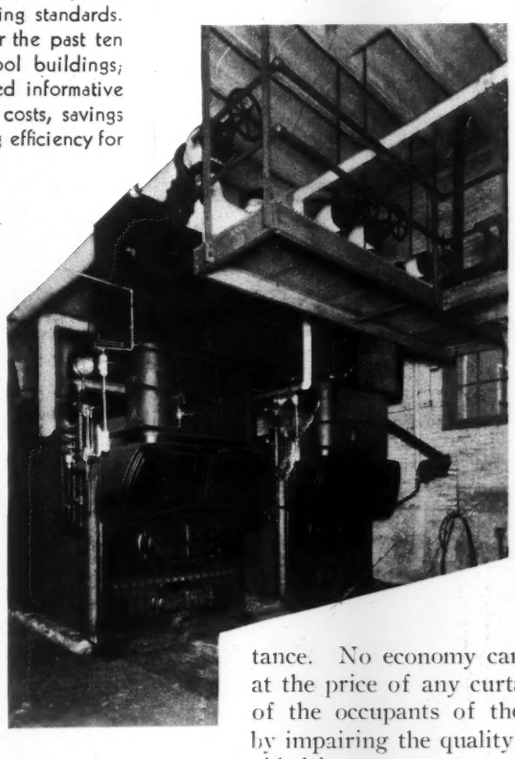
Vol. IV *January, 1932* **No. 1**

Promoting Operating Economy in School Plants

By John Howatt*

It is highly important—in these and all times—that public institutions and tax spending bodies achieve operating economies without lowering operating standards. Mr. Howatt has analyzed the records for the past ten years that have been kept on 400 school buildings; this article is the result. In it is included informative data on coal consumption, maintenance costs, savings that have been made, and how operating efficiency for a large school system can be obtained.



HOOKWAY ELEMENTARY SCHOOL BOILER PLANT CONSISTING OF TWO 155-HP. FIRE BOX DOWN DRAFT BOILERS

A GREAT deal of space in trade and technical journals and a great deal of time at meetings of engineering societies is given to discussions of problems of theory and design. Less space and time is given to consideration of the results obtained from the *application* of those theories and designs or to the *discussion of methods* that should be put into effect to bring about the highest possible operating efficiencies. The heating, straight ventilating or air conditioning plant designed so it can be operated at an efficiency of 75 per cent, but which is actually—either through neglect or ignorance—operating at an efficiency of 65 per cent, is not so good an investment as a plant designed and constructed so it can be operated at an efficiency of 70 per cent and is so operated.

Economies Must Not Impair Quality of Service

Increase in operating efficiencies, especially if accomplished without any increase in cost, is always desirable. Often, operating costs in school buildings may be reduced without curtailment in the service rendered. Insofar as problems of heating and ventilation are concerned, however, the health and comfort of pupils and teachers must be of paramount importance. No economy can be considered if it is obtained at the price of any curtailment in the health or comfort of the occupants of the building. Reducing coal bills by impairing the quality and quantity of ventilation provided is not economy. It is robbing the children in the classroom of that to which they are entitled.

First Step Toward Economy Is the Design

The first step toward economical school plant operation must be taken when the plant is being designed. A study of operating results should be the proper guide for the design and selection of equipment for new installations. It is often found that the operation of equipment over a term of years produces results different from that expected when the design was prepared. In selecting mechanical equipment, not only must the first cost be considered but the cost of main-

*Chief engineer, Bureau of Engineering, Board of Education, City of Chicago.

tenance and depreciation must be estimated and charged against the saving effected in using a type of "higher efficiency" apparatus. It may happen that due to the few hours of plant operation per year in school buildings, a high premium for efficiency may not be justified. This is in contrast to other types of buildings.

Fuel Consumption Records

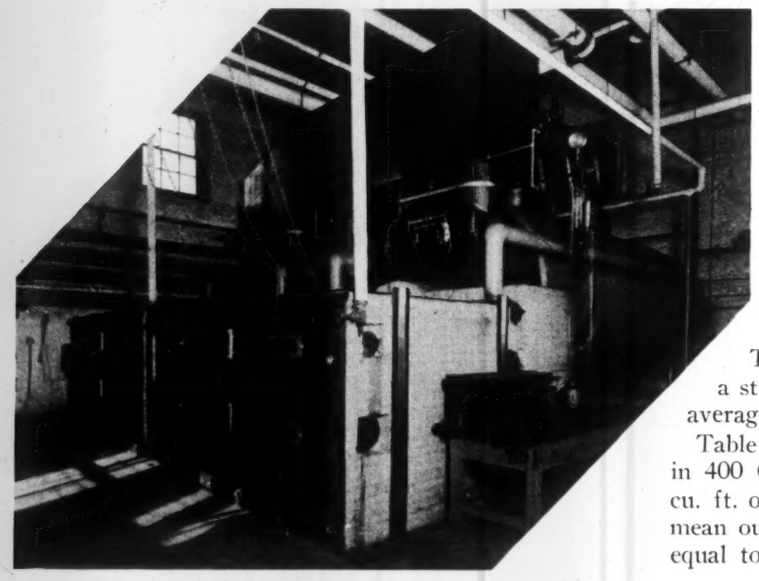
Records of the operation of heating and ventilating plants in 400 different public school buildings in Chicago have been kept for ten years. In these 400 school buildings can be found almost every conceivable type of heating and ventilating apparatus which has been designed within the past fifty years.

Fuel is one of the major operating costs; careful records of coal consumption in each of these 400 school buildings have been kept and attempts made to analyze the results. School buildings are classified in groups which are—as nearly as possible—similar in building design and in heating and ventilating equipment as well as use. It is useless to compare the coal consumption in one school building with another building of duplicate size, design, and equipment, unless consideration is given to the use of the building and its location as to weather exposures; for example, one building may be in a sheltered area and may be used almost exclusively for regular day school work with classes operated from 9 a.m. until 3:15 p.m. with no night activities, whereas a duplicate building may be located in an open and exposed area and used for frequent night activities or be run on an extended day school basis from 8 a.m. to 4 p.m.

After allowance has been made for all known factors, the differences in coal consumption in similar buildings are still great and can be accounted for only in the personnel of the teaching staff and of the operating crew. This is the factor that is beyond the control of the designing and construction engineers and is, to a considerable extent, beyond the control of the man in general charge of operations.

The Load on the Boiler Plant

The engineer designing the heating and ventilating plant for a school building must know the school load requirements before he can make any intelligent approach to the problem. It has been the writer's experience that in general the boiler plant in school buildings is of too large capacity for year-round operation. It is better to design the boiler plant so it will be over-loaded during the few sub-zero mornings in order that a better operating efficiency be maintained for the great part of the heating season. The largest load on a boiler plant in a school building in this climate comes, of course, in the early morning hours in heating up the building before the morning school session. A uniform temperature of 70 F is maintained by automatic heat regulation in all rooms in all school buildings in Chicago during class sessions. Because of the heat absorption of furniture and walls it is necessary to have the rooms up to temperature 45 minutes before the opening of school session on Monday morning and 30 minutes before the opening of school sessions on other mornings. Fig. 1 illustrates the boiler load at the different periods of the day as indicated by steam flow meters in the boiler plant of the Lucy Flower high school. Many such steam flow charts were obtained and the one illustrated in Fig. 1 was selected as typical, showing how rapidly the steam demand falls after 9 a.m. The minor waves in the load curve are caused by steam demands for heating water.



MARQUETTE ELEMENTARY SCHOOL BOILER PLANT CONSISTING OF TWO 175-HP. SCOTCH-MARINE BOILERS WITH DOWN DRAFT FURNACES

Table Gives Ten-Year Record of Coal Consumption

Inasmuch as there are so many factors affecting the coal consumption in the school buildings the study of one or two buildings might possibly lead to serious error in the interpretation of results. The greater the number of buildings included in a study of this sort, the less will be the error in the average results.

Table 1 gives a ten-year record of coal consumption in 400 Chicago public schools in lb. of coal per 1,000 cu. ft. of heated and ventilated space, together with the mean outdoor temperature for the heating season. Coal equal to coal mined in the State of Illinois was used

TABLE 1—TEN YEAR COAL CONSUMPTION RECORD, CHICAGO PUBLIC SCHOOLS

Average of All Buildings

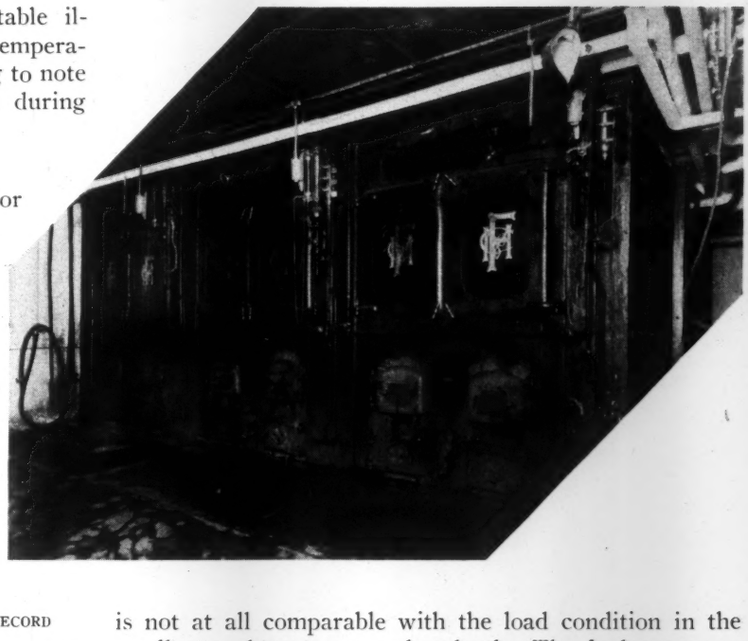
YEAR	(A)	(B)	(C)
1921	800	700	41.9°
1922	880	760	39.2°
1923	900	770	38.3°
1924	900	790	35.4°
1925	810	750	37.1°
1926	865	790	36.6°
1927	720	720	39.9°
1928	720	700	38.3°
1929	790	730	36.7°
1930	680	640	38.1°
AVERAGE	807	735	38.1°

A = Net pounds coal per 1000 cu. ft. in high schools.
B = Net pounds coal per 1000 cu. ft. in elementary schools.
C = Mean outdoor temperature during heating season.

throughout the entire ten-year period. The table illustrates how a slight change in outdoor mean temperature changes the coal consumption. It is pleasing to note the improvement in operating efficiency shown during the last four years.

Seasonal Consumption of Fuel

Table 2 illustrates the seasonal demand for steam in school buildings, showing the total tons of coal burned in 400 main public school buildings in Chicago during each of the seven winter months, from October 1 to April 30, during the years 1928-1930. Every system should maintain such a record in order to be able to anticipate the fuel demand for each month. The calculation of tons of coal burned per school day per degree the outdoor temperature was below 70 F is of interest.



McCosh Elementary School Boiler Plant Consisting of Three 66 In. by 18 In. H. R. T. Boilers

Table 2—Temperature and Coal Consumption Record

Month	Year	Mean Temp.	Tons Coal	Tons per Degree Below 70 F Per School Day
Oct.	1928	56.5	11,729	39.5
Nov.	1928	42.2	20,278	38.4
Dec.	1928	32.2	26,252	46.7
Jan.	1929	17.6	42,944	43.2
Feb.	1929	21.6	35,365	40.6
Mar.	1929	43.2	23,459	41.7
Apr.	1929	51.1	15,498	41.0
Oct.	1929	53.1	14,528	37.4
Nov.	1929	36.3	23,550	38.9
Dec.	1929	28.6	27,875	44.9
Jan.	1930	20.1	41,459	41.6
Feb.	1930	37.1	26,617	42.7
Mar.	1930	36.1	26,471	37.2
Apr.	1930	49.2	16,215	41.1
AVERAGE..	.....	37.5	25,160	41.07

Types of Boilers Used

The steam generating plant to be selected for a school building is, of course, one of the items which receives first consideration. About 900 boilers are operated in the main Chicago school buildings. A study of operating results in a large system must bring out interesting facts. In the large number of boiler plants under observation, the following groups are represented:

(1) Standard water tube boilers.
(2) Water tube boilers designed for U. S. Shipping Board during the war period.
(3) Standard Scotch-Marine boilers with corrugated furnaces.
(4) Ordinary horizontal return fire tube boilers with brick settings.
(5) Steel fire box boilers, portable, down draft, type.

The ordinary water tube type of boilers are installed principally in the larger high school buildings in which the load condition is not at all comparable with the load condition in the smaller or elementary grade schools. The fuel consumption per 1,000 cu. ft. of heated and ventilated space is much higher in this type of building than it is in a standard grade school because of the long school day and because of the demands for steam for many other purposes other than heating and ventilation.

Fuel Cost Is *One* Factor

All of the boilers included in the comparison have hand-fired furnaces. There are many stoker-fired plants in the Chicago public schools but these are not included in the averages under the different classifications. The fact that the fire box type of boiler plants are operated at low pressure aids in coal economy as it makes possible lower flue gas temperatures and lower radiation losses, but against this must be charged the cost of electric current for driving ventilating fans and other auxiliaries and difficulty in removing soot from flues, which may change the picture. The fuel cost of any plant is but one of the factors to be considered making up the total cost of operating boiler room equipment. The life of the boiler and apparatus, the freedom from

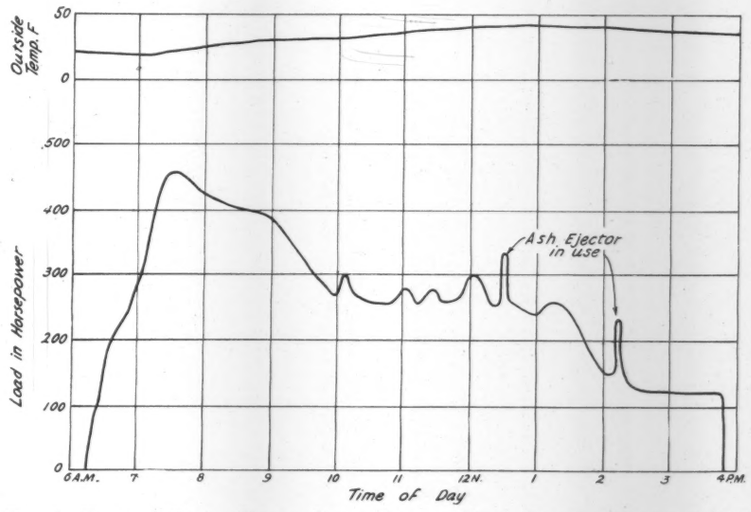
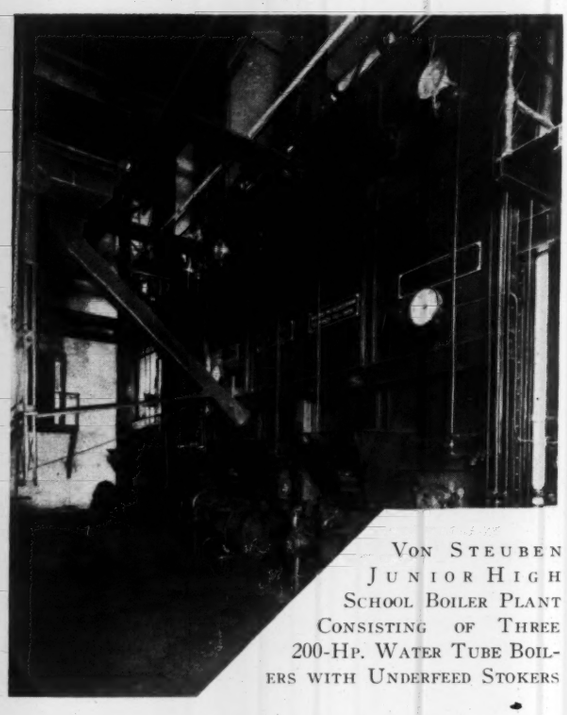


Fig. 1—Typical Daily Boiler Load, Flower High School, March 2, 1931



Von Steuben Junior High School Boiler Plant Consisting of Three 200-Hp. Water Tube Boilers with Underfeed Stokers

danger of failure during cold weather, reserve or overload possibility, and the cost of maintenance and repairs must all be considered.

The decision in the selection between the use of low pressure steam boiler plants and high pressure steam boiler plants in school work should be made always after a full consideration of the total estimated 25-year cost and other special considerations. The Chicago board of education, fortunately, has 22 elementary school buildings less than four years old of similar design and each containing about 1,000,000 cubic feet of heated and ventilated space which may be compared as to operating results. Some of these buildings are heated with low pressure steam plants using electric motors for driving the ventilating fans while others have boilers operated at 70-lb. steam pressure with steam engine-driven blowers. With electric power costing 3½c per kw-hr. and coal costing $4.60 per ton, the average results per school for the year July 1, 1930 to June 30, 1931 were as follows:

Low Pressure Plants		High Pressure Plants	
Coal cost	$1,055.00	Coal cost	$1,350.00
Power cost	580.00	Power cost	160.00
Total	$1,635.00	Total	$1,510.00

Cost of Furnace Maintenance in Chicago Schools

Furnace maintenance is the largest item about the boiler plant. The cost of fire brick repairs to the portable type fire box boiler is negligible. The average actual cost for the maintenance of the brick work and dutch oven type furnace used in front of the U. S. Shipping Board type of boiler in the Chicago public schools averages $1.25 per hp. per year. The average cost of fire brick repairs to the brick set horizontal return fire tube type of boiler with ordinary shaking grates is 60 cents per hp. per year. The average cost of maintenance for fire brick in the Scotch-marine type of boiler with down draft furnace in front has been 23 cents per hp. per year.

Study the Building

It has been determined that a factor of even more importance than the design of the plant, however, is the care and intelligence used in its operation. The building should be studied by the engineer in charge with the view of obtaining the maximum results with the minimum of effort and expense. A careful study of the building and its peculiarities or its characteristics will almost always disclose ways in which operating conditions can be improved at a reduction in operating costs. One factor that is of great importance, especially in cooling off the building at night, is the condition of the windows. We have records of school buildings where the average room temperature in the morning was 10 F higher after having windows caulked and weather-stripped; the entire cost of caulking window frames and weather-stripping windows was returned in the fuel saving of one year.

In fact the importance of keeping window frames and sash properly caulked and weather-stripped cannot be over-emphasized. Many classrooms found difficult to heat and made almost uninhabitable in inclement weather because of cold drafts are made comfortable by closing up the window cracks through which the cold air flows so freely. Money spent for caulking and weather-stripping is soon returned in the economies effected in fuel consumption and in the increased use and comfort of the building.

Cooling Off Rate

In considering the cooling off rate of buildings a great many cases must be considered to obtain dependable results; in a large system averages must be used in studying results. With the type of construction used in Chicago school buildings and the methods of operation of the plants, we find that during a period of nearly steady 30-F weather with no unusual high winds a school building left at 70 F at 3:15 p.m. on Friday will have an inside temperature of 45 F at 6 a.m. on Monday morning and 50 F on each morning of the week thereafter. When the fans are shut down and the supply of heat cut off, the temperature will fall from:

70 F to 63 F in the first 30 minutes
63 F to 61 F in the next 30 minutes
61 F to 60 F in the next 30 minutes
60 F to 59 F in the next 30 minutes

After that time the rate of cooling will become slower and slower. Wind causes the building to cool more rapidly than do low outdoor temperatures.

Effect of Drawing Shades Slight

The benefit of drawing down shades on windows as an aid in heating the buildings has been frequently discussed. Experiments were carried out for a period of two weeks in typical school buildings in Chicago last winter to learn just what was the effect of the shades on the heating and cooling of the room. It was found that the effect is so slight it is negligible in rooms heated by the air blast system. It is possible to heat up a room just a little faster in the morning with the shades down but the room cools off just as much over night and the whole effect is so slight as to be negligible. In a room heated by direct radiation, however, the advantage in drawing shades is more noticeable.

The Question of Banking Fires

In plants such as school buildings, where the heating plant is shut down, the question as to whether or not fires should be banked in the boiler plant is one which is frequently raised. A study of this matter was made in the Chicago public schools with the conclusion that there is little choice between banking the fires at night or burning them out and starting a fresh fire every morning. Where banking is practiced it should be done with as small an amount of coal as will carry over the fire until the next morning, the best method of banking of different types of furnaces to be determined by the operating engineer by trial, depending upon the type of furnace and the kind and quality of fuel available. When the fires are not banked the ashes from the burnt out coal should be left on the grates over night to help check the flow of air through the furnace and boilers and prevent excessive cooling. Steam can be raised more quickly starting from a fresh fire than from a banked fire but the time required to build a fresh fire will make up the difference in hours of labor involved.

Standardized Instructions Promote Economy

Where there are a large number of buildings under one organization, general instructions should be standardized as far as possible. Instructions to engineers in charge of school buildings should include the following points:

(1) Keep the heat regulation system in good condition. Overheating of rooms is unhealthful and wasteful.

(2) Set vent dampers on the ventilating system ducts so a slight pressure will be built up in the building to prevent drafts and excessive air infiltration.

(3) Adjust the roof dampers on the vents so they can be closed tightly. Close these dampers fifteen minutes before shutting down the air delivery fans to build up excess pressure in the building and prevent cooling off of the building due to the opening of main exit doors.

(4) Heat the building to the desired temperature only early enough before class sessions commence to bring the building, walls, and furniture up to temperature.

(5) Keep valves and steam traps in good repair so all leaks and wastage are stopped.

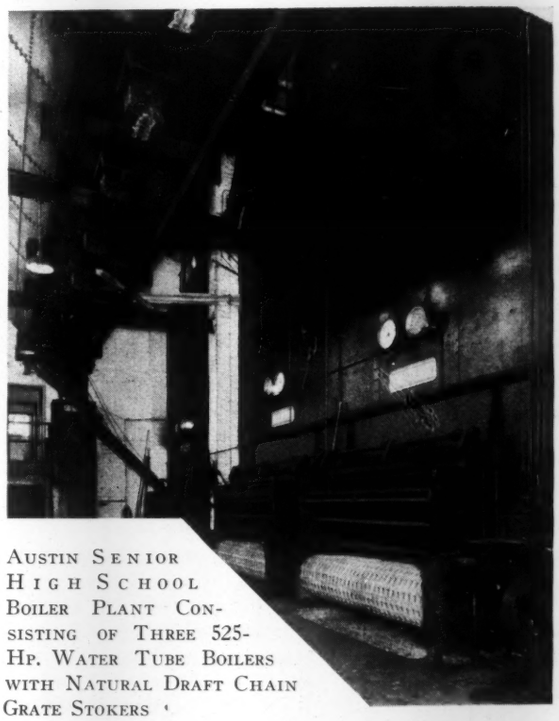
(6) Keep the relative humidity at 40 per cent in order to provide comfort to the pupils and teachers with a temperature of 70 F.

(7) Keep windows caulked and weather-stripped.

(8) Utilize as much of the heating system as is needed to heat up quickly in the morning, thus saving time and fuel. Cut out boilers as soon as the peak load is over. Do not fire two boilers when one will carry the load.

(9) The fireman on the job is a most important factor in the fuel economy. Spend time teaching him to do his work in the most efficient way.

(10) Some men seem to think the boiler damper has but two positions, fully open and fully closed. Sufficient draft is provided to burn coal rapidly enough to take care of the load requirements during —10 F weather. As soon as the peak load is over, which usually is within three hours after the steam is turned on, boiler dampers should be partly closed to prevent excess heat waste up the stack.



Austin Senior High School Boiler Plant Consisting of Three 525-Hp. Water Tube Boilers with Natural Draft Chain Grate Stokers

(11) Dirty boilers are a common cause of fuel waste. With the smoky bituminous coal generally burned in middle west plants soot deposits form rapidly. This is especially true in the water tube type of boiler. Soot must be thoroughly removed daily. Scale formation is less troublesome in school work because such a large percentage of the water fed back to the boiler is condensate; however, formation of scale also must be watched.

(12) Care must be exercised to see that all of the available heat in the furnace is used up when the plant is to be shut down for the day.

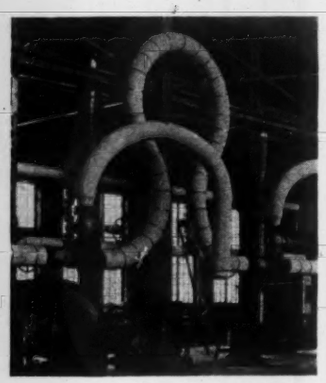
(13) Smaller sizes of coal can usually be burned efficiently. Break up large lumps before firing. Watch for clinkers and holes in the fire as they hinder combustion of gases. Avoid excessive stirring up of the fuel bed.

(14) Keep the boiler setting tight. Check up occasionally with draft and CO_2 readings.

(15) Practice recirculation of air in the morning and at all other times when the building is not occupied but when it must be heated, in order to conserve fuel.

(16) The engineer must take pride in the condition and appearance of the plant under his charge. The equipment should look well and function well. It should be kept in proper repair and not permitted to fail because of lack of minor repair. Economical operation of the school plant depends largely upon the person in charge of that plant. The difference in rates of coal consumption in similar buildings and in the operating costs in similar buildings is due to the personnel of the operating force. It is found that plant economies will vary in proportion to the ability and sense of duty of the one in charge at each building.

Editor's Note: In the article "Fuel Consumption in Toledo Schools," Samuel R. Lewis provided similar information for another extensive system. This article was published in the December, 1930, Heating, Piping, and Air Conditioning, page 997.



Method and Charts Economical Thickness

By Temple C. Patton*

FREQUENTLY the engineer is confronted with the problem of determining what thickness of insulation covering will give the most economical results. It is obvious that beyond a certain thickness, the saving due to further insulation covering is not sufficient to offset the added cost of its application. For ordinary conditions the value of the correct thickness is fairly well worked out, but in many cases conditions arise where it is difficult to determine the most economical amount.

The following method is submitted as a basis by means of which an engineer can derive an economical thickness.

Assumptions Made

One assumption is made which is approximately correct and should not affect the precision of the final result by more than 5 per cent. This is the cost of the insulation material plus its cost of application, which is assumed to be directly proportional to the number of board feet of insulation material used.

The method employed in these derivations is in no way dependent upon the heat loss from the bare surface to be insulated, but it does demand that the temperature of the inside insulation surface remain constant as varying amounts of insulation are applied outside. This demand is met exactly in pipe lines which are used to convey saturated vapors such as the ordinary low pressure steam line. For a number of other conditions this constancy of inside insulation surface temperature is closely obtained, as in a temperature-controlled furnace.

Three Type Cases Dealt with

The following article deals with three type cases which are regularly met with:

One insulation material applied to a flat surface.
Two insulation materials applied to a flat surface.
One insulation material applied to circular piping.

The solution of any one of these cases is dependent upon the following controlling factors, each of which must be first definitely determined:

a. Coefficient of conductivity of the insulation material (K) expressed as Btu, hour $^{-1}$, ft. $^{-2}$, F $^{-1}$, inch.
b. Emission coefficient of the insulation material (E) expressed as Btu, hour $^{-1}$, ft. $^{-2}$, F $^{-1}$.
c. Cost per board foot of insulation material plus cost of application (c) expressed as dollars per board foot.
d. Annual fixed charges as applied against cost of insulation material plus cost of application ($F.C.$) expressed as a percentage (%).
e. Hours per year that the insulation material is in actual use (Y) expressed as hours of operation.
f. Temperature of the surface to be insulated (t_h) expressed in degrees Fahrenheit (F); assumed to be constant.
g. Outside room or air temperature (t_r) expressed in degrees Fahrenheit (F); assumed to be constant.
h. Cost of one million useful Btu (C) expressed as dollars per one million (1,000,000) Btu.

Other additional data necessary for certain type cases will be noted as the cases are taken up.

One Insulation Material Applied to a Flat Surface

The first and simplest type case consists in determining the economical thickness when one insulation material is applied to a flat surface. A general equation is first set up for the yearly net saving expressed as dollars per year and based on one square foot of surface.

$$\text{Net Saving} = \left[\begin{array}{c}\text{Original loss}\\ \text{with no}\\ \text{insulation}\end{array}\right] - \left[\begin{array}{c}\text{Loss with } (th)\\ \text{inches of}\\ \text{applied}\\ \text{insulation}\end{array}\right] - \left[\begin{array}{c}\text{Annual fixed}\\ \text{charges as ap-}\\ \text{plied against}\\ \text{the cost of the}\\ (th) \text{ thick ma-}\\ \text{terial plus the}\\ \text{cost of its ap-}\\ \text{plication.}\end{array}\right]$$

or expressed more explicitly in terms of the nomenclature noted above,

$$\text{Net Saving} = \left[\begin{array}{c}\text{Original loss}\\ \text{with no}\\ \text{insulation}\end{array}\right] - \left[\frac{Y \times C \times \Delta t \times K}{1{,}000{,}000 \times (th + a)}\right] - \left[F.C. \times th \times c\right]$$

where $a = K/E$ and $\Delta t = t_h - t_r$

To determine the thickness which will give the maximum net saving, the net saving as thus expressed is differentiated with respect to thickness and the resultant expression equated to zero.

$$\frac{d\ (\text{Net Saving})}{d\ (th)} = \left[\frac{Y \times C \times \Delta t \times K}{10^6\ (th + a)^2}\right] - \left[F.C. \times c\right] = 0$$

If the expression is rearranged to obtain the thickness as an explicit function of the other factors, there is obtained directly a value of the thickness which makes the net saving a maximum.

$$\text{Economical thickness} = \sqrt{\frac{Y \times C \times \Delta t \times K}{F.C. \times c \times 10^6}} - \frac{K}{E}$$

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For Determining of Insulation

This article—and the charts published with it—gives the engineer a method of arriving at an economical thickness of insulation for a large number of conditions which are actually met with, in an orderly fashion and not merely by hit-or-miss methods or guess-work. The method is independent of whatever heat loss may have been in effect previous to the application of the insulating material.

Three type cases are taken up: (1) The case of one insulation material applied to a flat surface; (2) the case of two insulation materials applied to a flat surface; and (3) one insulation material applied to piping.

The scope of the article is carefully outlined, as is the method of using the information which is presented.

Example Illustrates Problem

An example is submitted to illustrate an actual problem:

$K = 0.6$
$E = 2.0$
$\Delta t = 500.0$
$C = 0.48$
$Y = 7300.0$
$F.C. = 0.15$
$c = 0.28$

$$\text{Economical thickness} = \sqrt{\frac{7300 \times 0.48 \times 500 \times 0.6}{0.15 \times 0.28 \times 10^6}} - \frac{0.6}{2} = 5 - 0.3 = 4.7 \text{ in.}$$

Theoretically a thickness of 4.7 in. is called for but in actual practice the nearest smaller commercial size would be used.

Two Insulation Materials Applied to a Flat Surface

The second type case, dealing again with the application of insulation to a flat surface, is complicated by the use of two materials, one over the other.

The outer covering is material of high insulation value but its use is limited by the temperature (t_m) at which physical or chemical deterioration of the material occurs. The inner covering is made of a more refractory insulation material which can be subjected to greater temperature but which has a lower insulation value.

The problem consists (1) in applying an inner covering of such thickness that the outside surface temperature of this covering is safe for the application of the outer covering, and (2) in making the thickness of each covering the most economical commensurate with condition (1).

Obviously, it will be most practicable to apply the outer covering just as soon as the temperature of the inner covering will permit. A solution of this problem requires data additional to those noted in the previous case, viz:

i. Temperature at which physical or chemical deterioration of the outer material occurs (t_m), expressed in degrees Fahrenheit. This temperature will be the temperature at the interface of the two materials under operating conditions.

j. Δt_o = temperature drop through outer covering.

$\Delta t_i = t_h - t_m$ = temperature drop through inner covering.

$\Delta t'_o = t_m - t_r$ = temperature drop through outer covering plus temperature drop from outer covering surface to air.

The subscripts (i) and (o) will be used to designate the constants or values of the inner and outer insulation coverings respectively.

Again a general equation is set up for the yearly net saving expressed as dollars per year and based on one square foot of surface.

$$\text{Net Saving} = \left[\begin{array}{c}\text{Original loss}\\ \text{from}\\ \text{uninsulated}\\ \text{flat surface}\end{array}\right] - \left[\frac{Y \times C}{10^6} \times \left(\frac{\Delta t}{\frac{th_i}{K_i} + \frac{th_o}{K_o} + \frac{1}{E_o}}\right)\right] - F.C.\left[c_i\, th_i + c_o\, th_o\right]$$

Let $a = K_o/E_o$

The expression in the parenthesis above can be brought into the form below by substituting $a = K_o/E_o$:

$$\left(\frac{\Delta t}{\frac{th_i}{K_i} + \frac{(th_o + a)}{K_o}}\right)$$

The term involving this expression is next dealt with separately. Since the number of Btu's coming through the inner covering is the same as the number going through the outer covering into the air, the equality below is valid.

$$\frac{\Delta t_i K_i}{th_i} = \frac{\Delta t'_o K_o}{(th_o + a)} = \frac{\Delta t'_o K_o}{th'_o}, \quad where \quad (th_o + a) = th'_o$$

or rearranging to obtain th_i in terms of th'_o:

$$th_i = \Delta t_i / \Delta t'_o \times K_i / K_o \times th'_o$$

Substituting this value of th_i and simplifying:

$$\left[\frac{Y \times C}{10^6} \times \frac{\Delta t}{\frac{\Delta t_i}{\Delta'_o} \times \frac{K_i}{K_o} \times \frac{th'_o}{K_i} + \frac{th'_o}{K_o}}\right] = \left[\frac{Y \times C}{10^6} \times \frac{\Delta t}{\left[\frac{th'_o}{K_o}\right]\left[1 + \frac{\Delta t_i}{\Delta t_o}\right]}\right] = \left[\frac{Y \times C}{10^6} \times \frac{\Delta t}{\frac{th'_o}{K_o} \times \frac{\Delta t}{\Delta t'_o}}\right] = \left[\frac{Y \times C}{10^6} \times \frac{\Delta t'_o \times K_o}{th'_o}\right]$$

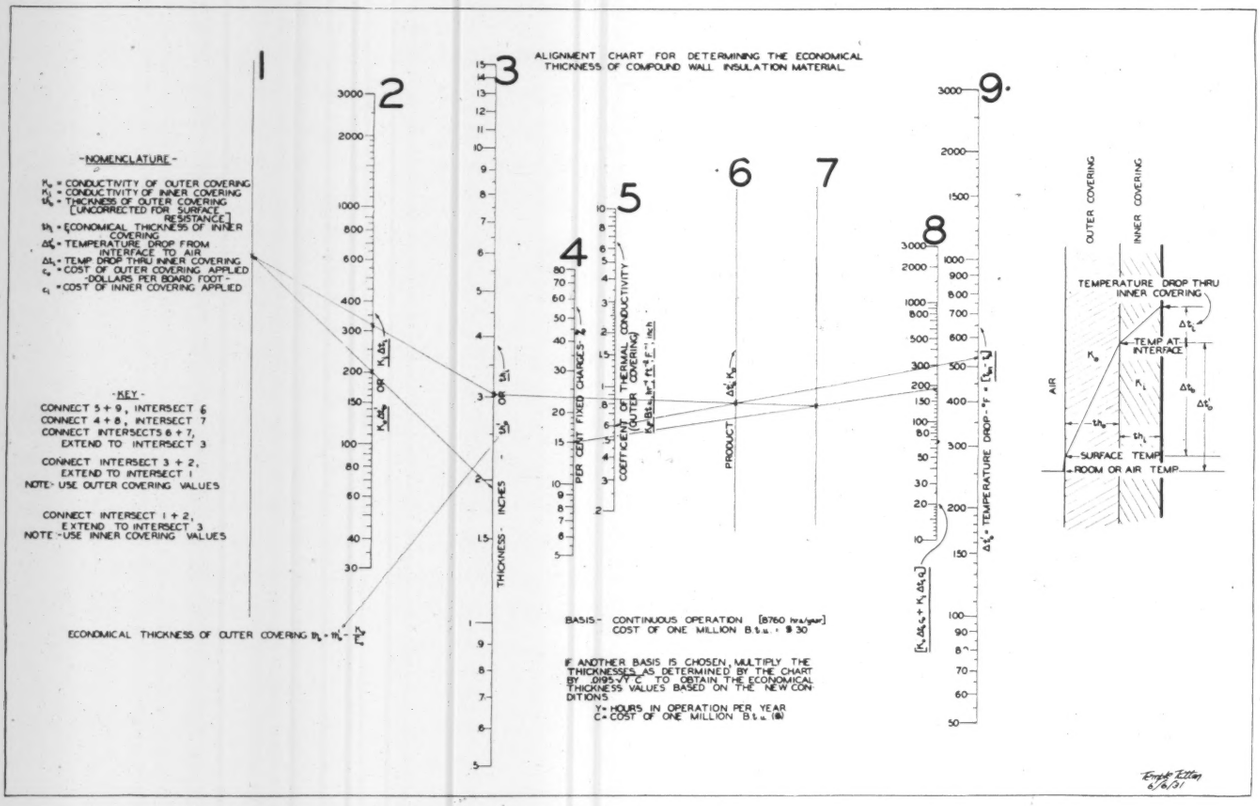

ALIGNMENT CHART FOR DETERMINING THE ECONOMICAL THICKNESS OF COMPOUND WALL INSULATION MATERIAL
-NOMENCLATURE-
-KEY-
CONNECT 5 + 9, INTERSECT 6
CONNECT 4 + 8, INTERSECT 7
CONNECT INTERSECTS 8 + 7, EXTEND TO INTERSECT 3
CONNECT INTERSECT 3 + 2, EXTEND TO INTERSECT 1
NOTE- USE OUTER COVERING VALUES
CONNECT INTERSECT 1 + 2, EXTEND TO INTERSECT 3
NOTE -USE INNER COVERING VALUES
ECONOMICAL THICKNESS OF OUTER COVERING
PER CENT FIXED CHARGES
COEFFICIENT OF THERMAL CONDUCTIVITY (OUTER COVERING)
PRODUCT
THICKNESS - INCHES
TEMPERATURE DROP THRU INNER COVERING
TEMP AT INTERFACE
SURFACE TEMP.
ROOM OR AIR TEMP.
AIR
OUTER COVERING
INNER COVERING
BASIS:- CONTINUOUS OPERATION [8760 hrs/year]
COST OF ONE MILLION B.t.u. = $ 30
IF ANOTHER BASIS IS CHOSEN, MULTIPLY THE THICKNESSES AS DETERMINED BY THE CHART BY .0195 √Y C TO OBTAIN THE ECONOMICAL THICKNESS VALUES BASED ON THE NEW CONDITIONS
Y = HOURS IN OPERATION PER YEAR
C = COST OF ONE MILLION B.t.u.

Fig. 1—By means of this chart, the economical thickness of compound wall insulation material can be determined

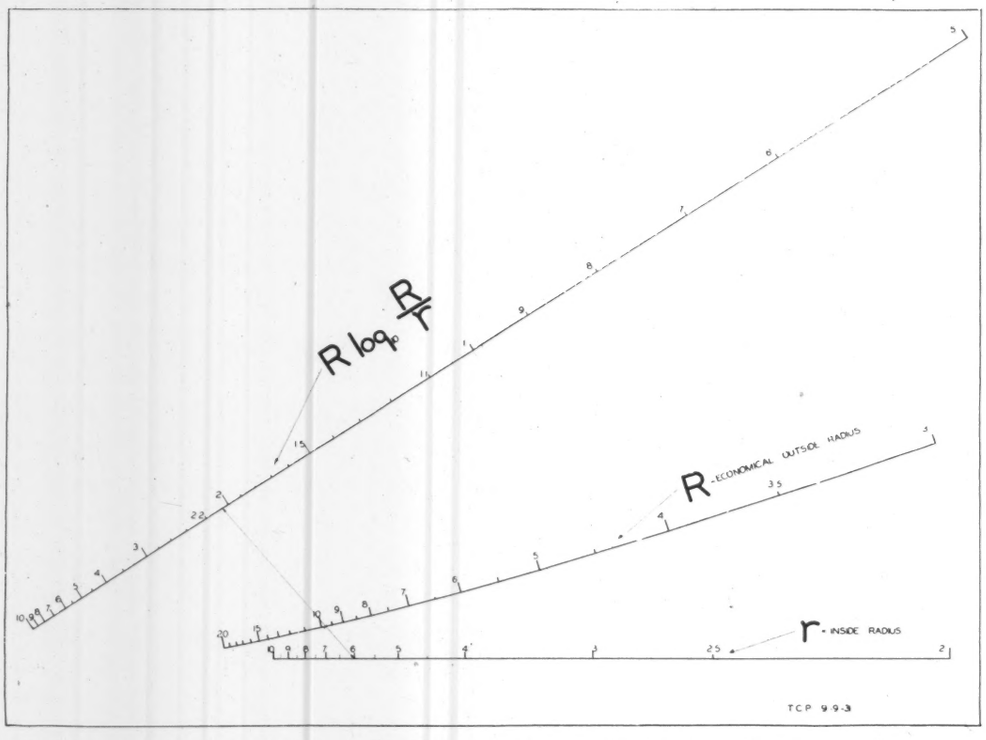


Fig. 2—In determining the economical thickness of insulation applied to piping, this chart obviates the tedious process of obtaining R, (the outside radius) by successive approximations

The general equation for the net saving now assumes this form:

$$\text{Net Saving} = \left[\begin{matrix}\text{Original loss}\\ \text{from}\\ \text{uninsulated}\\ \text{flat surface}\end{matrix}\right] - \left[\frac{Y \times C}{10^6} \times \frac{\Delta t'_o \times K_o}{th'_o}\right] - F.C.\left[\frac{\Delta t_1 \times K_1 \times th'_o \times c_1}{\Delta t'_o \times K_o} + c_o th_o\right]$$

To find the correct thickness which will make the net saving a maximum, the net saving as thus expressed is differentiated with respect to the thickness of the outer covering and the resultant expression equated to zero.

$$\frac{d\,(\text{Net Saving})}{d\,(th_o)} = \frac{Y \times C \times \Delta t'_o \times K_o}{10^6}\left(\frac{1}{th'_o}\right)^2 - F.C.\left[\frac{\Delta t_1 \times K_1 \times c_1}{\Delta t'_o \times K_o} - c_o\right] = 0$$

Simplifying and obtaining th_o in terms of the other factors:

$$\text{Economical } th_o = \Delta t'_o \times K_o \sqrt{\frac{Y \times C}{10^6 \times F.C.\,[\Delta t_1 K_1 c_1 + \Delta t'_o K_o c_o]}} - \frac{K_o}{E_o}$$

The inner economical thickness is given from the following equality:

$$th_1 = \frac{\Delta t_1}{\Delta t'_o} \times \frac{K_1}{K_o} \times \left(th_o + \frac{K_o}{E_o}\right)$$

Example Illustrates Problem

An example is submitted to illustrate an actual problem.

$c_1 = 0.40$ $\quad Y = 4380$ (half-time operation)
$K_o = 0.60$ $\quad t_m = 600$ F
$E_o = 2.0$ $\quad t_h = 800$ F
$K_1 = 1.0$ $\quad t_r = 70$ F
$c_o = 0.35$ $\quad C = 0.30$
$F.C. = 0.15$

$\Delta t'_o = t_m - t_r = 600° - 70° = 530°$
$\Delta t_1 = t_h - t_m = 800° - 600° = 200°$

Economical outside thickness:

$$th_o = 530° \times 0.60 \sqrt{\frac{4380 \times 0.30}{10^6 \times 0.15 \times [200 \times 1.0 \times 0.40 + 530 \times 0.6 \times 0.35]}} - \frac{0.6}{2.0}$$

$$th_o = 318 \times \sqrt{\frac{1320.}{0.15 \times 10^6\,[80 + 110]}} - 0.3$$

$$th_o = 0.318 \sqrt{46.3} - 0.3 = 2.16 - 0.30 = 1.86 \text{ in.}$$

Economical inside thickness:

$$\frac{200°}{530°} \times \frac{1}{0.6} \times (1.86 + \frac{0.6}{2}) = 1.36 \text{ in.}$$

Use of Chart Solves This Case Quickly

By the use of the alignment chart an alternate, quick, and accurate graphical solution may be obtained of this type case. As before, all the constants of the insulation materials must first be determined.

After definite values have been assigned to these items, values for simple combinations of the above basic factors are evaluated as below:

(a) $K_o \Delta t'_o$
(b) $K_1 \Delta t'_1$
(c) $(\Delta t_1 K_1 c_1 + \Delta t'_o K_o c_o)$
(d) $t_h - t_m = \Delta t_1$
(e) $t_m - t_r = \Delta t'_o$
(f) K_o / E_o
(g) $0.0195 \sqrt{Y \times C}$

The immediate solution for the economical thicknesses of the insulation materials to be applied can now be obtained by connecting the values on the various scales by straight lines, as explained by the key of the chart. (Fig. 1.)

Following the procedure as directed by the chart key, a thickness of the outer layer is first determined (scale *3* of the chart), but this is an uncorrected thickness and must be corrected for surface emission resistance. The final corrected outer economical thickness for actual use is obtained by subtracting the value of the expression K_o/E_o from the uncorrected outer thickness value.

The thickness of the inner layer is next determined as directed by the chart key. Note in obtaining the intersection on line *1* that the value to be used on the *3*-scale is the uncorrected outer thickness value; the value on the *2*-scale the product of $(K_o \Delta t'_o)$.

The intersection on line *1* now determines the starting point of a straight line which when passed through the value of $(K_1 \Delta t_1)$ as found on scale *2*, will intersect scale *3* at the correct value for the most economical inner thickness of material.

How the Chart Is Used

The outline preceding is a generalized system of procedure. A specific problem is offered as an opportunity to run through a practical example. Values for the controlling factors which determine the solution are given below and approximate a set of conditions that one might expect to encounter.

$K_o = 0.6$ $\quad Y = 4380$ (half-time operation)
$E_o = 2.0$ $\quad t_m = 600$ F
$K_1 = 1.0$ $\quad t_h = 800$ F
$c_o = 0.35$ $\quad t_r = 70$ F
$c_1 = 0.40$ $\quad C = 0.30$
$F.C. = 0.15$

Values for simple combinations of the above are next evaluated as previously outlined.

$\Delta t_1 = t_h - t_m = 200°$
$\Delta t'_o = t_m - t_r = 530°$
$K_o \Delta t_o = 318$
$K_1 \Delta t_1 = 200$
$K_o / E_o = 0.30$
$(\Delta t'_o K_o c_o + \Delta t_1 K_1 c_1) = 190$
$0.0195 \sqrt{Y \times C} = 0.707$

Solving by the chart, we arrive at a value of 3.04 in. for the outer uncorrected thickness. Continuing the solution by the chart, the value of 1.91 in. is obtained for the inner economical thickness.

The above thicknesses are based on *full time operation*. To determine the economical thicknesses for half-time operation or 4,380 hours per year, multiply the thicknesses as derived above by the value of the expression $0.0195 \sqrt{Y \times C}$, or 0.707 *in this specific problem*, which will give the economical thicknesses for half-time operation.

Since the values obtained are probably thicknesses which do not correspond to commercial sizes, it is most important that the proper nearest commercial size be chosen for use. A general rule is always to pick a commercial thickness which is less than the economical thickness for the outer covering, but greater than the inner economical thickness for the inner covering.

Converting Thicknesses for Full-time Operation to Half-time Operation

Following is the method for converting the thicknesses from a full time yearly basis to half-time operation:

$th'_o = 0.707 \times 3.04 = 2.15$ in. (uncorrected outside economical thickness for half-time operation)

Note that in making this conversion the *uncorrected* outer thickness value is used.

Diminishing this value by the value of the expression $K_o/E_o = 0.3$ gives a correct economical thickness of 1.85 in. for the outer covering.

$$th_o = 2.15 - 0.30 = 1.85 \text{ in.}$$

The correct inside economical thickness is obtained directly and needs no correction for surface resistance.

$$th_i = 0.707 \times 1.91 = 1.35 \text{ in.}$$

	Theoretical thickness	*Commercial thickness*
Outside covering	1.85 in.	1.5 in.
Inside covering	1.35 in.	1.5 in.

Insulation Applied to Piping

The third type case deals with one kind of insulation material applied to circular piping. This problem is slightly more involved because of the introduction of log mean area.

As in the previous cases, an equation for the yearly net saving is set up, but the basis chosen in this case is one foot of pipe length. The letters R and r expressed in inches are used to denote the outside and inside radii respectively.

R = outside radius
r = inside radius

$$\text{log mean radius} = \frac{R - r}{ln_e \frac{R}{r}}$$

$$\text{Net Saving} = \left[\text{Original loss from bare piping}\right] - \left[\frac{Y \times C \times \Delta t \times 2\pi}{10^6\left[\frac{12 ln_e R/r}{K} + \frac{12}{ER}\right]}\right] - F.C. \times c\,[\pi/12]\,(R^2 - r^2)$$

Let $a = K/E$

$$\text{Net Saving} = \left[\text{Original loss}\right] - \left[\frac{Y \times C \times \Delta t \times 2\pi \times 10^{-6} \times K}{12\left[ln_e \frac{R}{r} + \frac{A}{R}\right]}\right] - F.C. \times c \times \frac{\pi}{12}(R^2 - r^2)$$

Differentiating the net saving with respect to the outside radius R:

$$\frac{d\,(\text{Net Saving})}{d\,(R)} = [Y \times C \times \Delta t \times 10^6 \times K \times \pi/6]\left[\frac{1}{\left(ln_e \frac{R}{r} + \frac{a}{R}\right)^2}\right]\left[\frac{1}{R} - \frac{a}{R^2}\right] - [F.C. \times c \times \pi/12 \times 2R] = 0$$

By rearranging terms the expression can be brought into the following form:

$$\sqrt{\frac{Y \times C \times \Delta t \times 10^6 \times K}{F.C. \times c}} = \left(Rln_e \frac{R}{r} + a\right)\sqrt{\frac{R}{R - a}}$$

The left-hand side of this equality can be directly evaluated but the determination of what value of R (the most economical R) will make the right-hand term equal to it, requires an indirect method.

To obviate the laborious and tedious process of obtaining R by successive approximations, the following shorter method is suggested in conjunction with the chart shown as Fig. 2, (for pipe *radii* of 2 in. or more).

When the value of K/E is 0.3, it introduces an error of about 8 per cent when dealing with a 2-in. *radius* pipe, an error of 5 per cent when dealing with a 3-in. *radius* pipe, and smaller percentage errors with larger piping. In case greater accuracy is desired, the value as derived can be used as a close approximation to an exact value, where an exact value is to be obtained by successive approximations.

The shorter method assumes the value of $\sqrt{\frac{R}{R - a}}$ as equal to 1, which simplifies the expression above to:

$$\sqrt{\frac{Y \times C \times \Delta t \times 10^{-6} \times K}{F.C. \times c}} - a = Rln_e \frac{R}{r}$$

It is more convenient to work with logarithms to a base of *ten* instead of e and this change is noted below:

$$\frac{1}{2.3}\left(\sqrt{\frac{Y \times C \times \Delta t \times 10^{-6} \times K}{F.C. \times c}} - \frac{K}{E}\right) = R\ log_{10} \frac{R}{r}$$

The left-hand side of this expression is still capable of direct evaluation, but the value of R must be indirectly obtained. Fig. 2 is designed to do this directly for the range of R-values ordinarily met.

The key of the chart simply requires connecting the value of the left side of the equation, as located on its scale, with the value of the inside *radius,* as located on its scale. The value at the intersection of this straight line with the R-scale directly gives the most economical outside *radius* to use.

Example

$K = 0.6$
$E = 2.0$
$\Delta t = 500$ F
$c = 0.28$
$Y = 7300$
$F.C. = 0.15$
$C = 0.48$

$$\frac{1}{2.3}\left(\sqrt{\frac{7300 \times 0.48 \times 500 \times 0.6}{0.15 \times .28 \times 10^6}} - 0.3\right) = R\ log_{10}\ R/r$$

$$\frac{1}{2.3}\ (5 - 0.3) = 2.04 = R \log_{10} R/r$$

Three pipe sizes are considered to show the direction and magnitude of change of the economical thickness of covering with pipe radius. The economical thickness is obtained by subtracting the inside radius from the outside economical radius.

$$2.04 = R\ \log_{10} \frac{R}{r}$$

The value of 2.04 is located on its own scale and is successively connected with three values of the inside radii. The values at the intersection of these straight lines with the R-scale are noted in the table below:

Inside radius r	*Economical outside Radius R*	*Economical thickness R—r*
2 in.	5.1 in.	3.1 in.
6	9.8	3.8
10	14.0	4.0
Infinity	——	4.7

It is interesting to note that when the inside radius has the dimension of infinity, then there is obtained an essentially flat surface and the first type case is duplicated. The economical thickness as derived for a flat surface previously gave a result of 4.7 in., as noted above.

Scope of This Method

It is evident that the scope of the foregoing article is limited; first, because of the assumption that the cost of material plus its cost of application is proportional to the number of board feet of insulation material applied, and second, because the temperature of the surface to be insulated is considered constant as varying thicknesses of insulation material are applied, but these fixed conditions in no way defeat the purpose of the article, for it gives the engineer a method of arriving at an economical thickness for a large number of conditions which are actually met with, in an orderly fashion and not merely by hit-or-miss methods or by guess-work.

Furthermore, this method has the distinct advantage of being usable independent of whatever heat loss may have been in effect previous to the application of the material.

The material is arranged to enable the engineer to determine quickly an economical thickness without involving himself too deeply in mathematical computations. With a little study of the method employed and the attending examples, situations not covered by this present article can be more confidently studied.

Summary of Formulas

Flat Surface—one material:

$$\text{Economical thickness} = \sqrt{\frac{Y \times C \times \Delta t \times K}{F.C. \times c \times 10^6}} - \frac{K}{E}$$

Flat Surface—two materials:

Economical outside thickness =

$$\Delta t_o' \times K_o \sqrt{\frac{Y \times C}{10^6 \times F.C. \, (\Delta t_1 K_1 c_1 + \Delta t'_o K_o c_o)}} - \frac{K_o}{E_o}$$

$$\text{Economical inside thickness} = \frac{\Delta t_1}{\Delta t'_o} \times \frac{K_1}{K_o} \times \left(th_o + \frac{K_o}{E_o} \right)$$

Circular piping—one material:

Economical thickness = Economical $R—r$
where R is obtained from the equation below,—

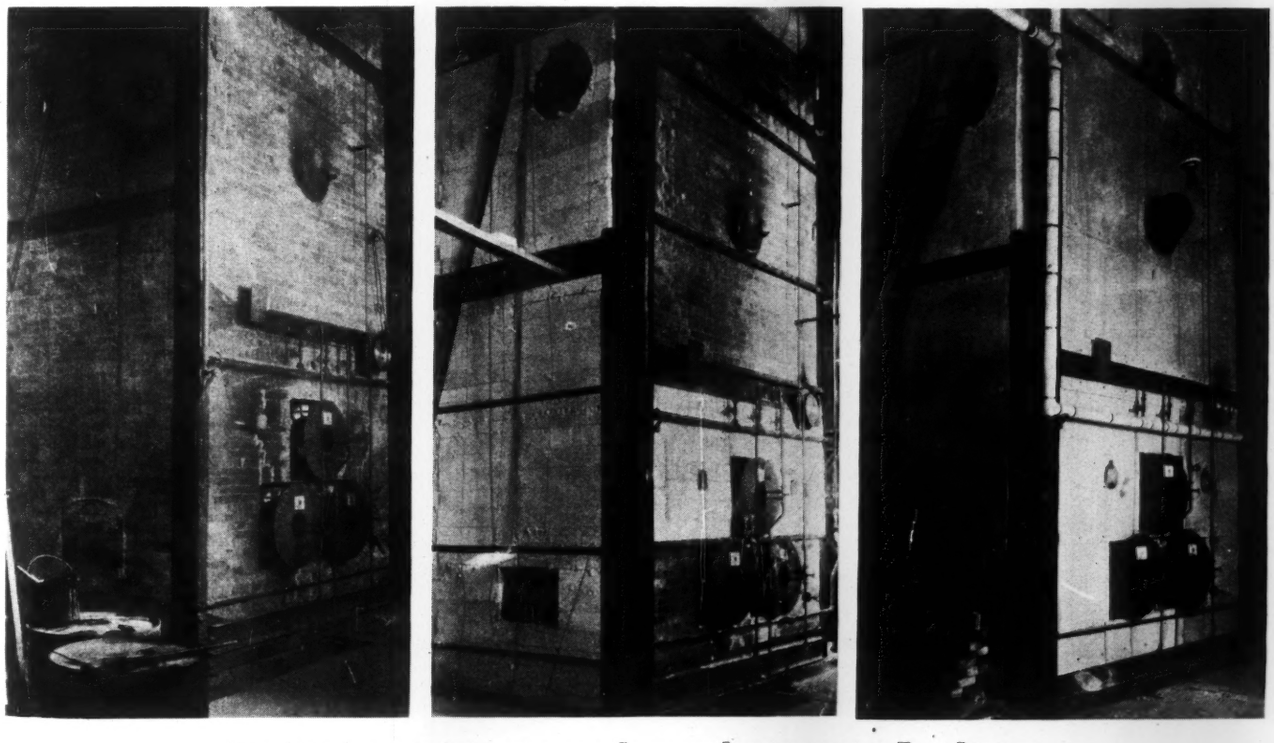
Original derivation:

$$\sqrt{\frac{Y \times C \times \Delta t \times 10^{-6} \times K}{F.C. \times c}} = \left(R \ln_e \frac{R}{r} + \frac{K}{E} \right) \sqrt{\frac{R}{R—a}}$$

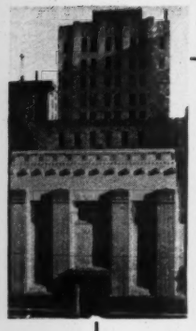
Simplified equation:

$$\frac{1}{2.3} \left(\sqrt{\frac{Y \times C \times \Delta t \times 10^{-6} \times K}{F.C. \times c}} - \frac{K}{E} \right) = R \log_{10} \frac{R}{r}$$

The author wishes to express his indebtedness to Professor Gordon B. Wilkes, Massachusetts Institute of Technology, for valuable suggestions and also to the Babcock and Wilcox Company for their permission to include in this article some material originally prepared for their private use.



THREE STEPS IN THE APPLICATION OF A COMPOUND INSULATION TO A FLAT SURFACE



This article includes information on
Zoning
Air Conditioning
Location of Equipment
Control
Recirculation
Quantities of Air
Cost of Power
Cost of Installation
—and describes a typical installation

By Ralph E. Phillips*
and
Donald T. Robbins†

Lay-Out of a Zoned Air Conditioning System

THE most important problems to be met in the construction of the new Los Angeles Stock Exchange building were those pertaining to the mechanical devices for trading facilities, proper illumination and proper air conditioning.

Air Conditioning a Paramount Requirement

The site of the building is an interior lot 80 ft. wide by 150 ft. deep. All of the mechanical spaces, store rooms and safe deposit vaults are in the basement, the entrance lobby, Western Union, Federal and Postal Telegraph offices are on the ground floor, and the second floor is devoted entirely to the main trading room, members' lounge and comparison room; the trading room extends in height through three floors. The fifth floor is devoted to the clearing house and its allied departments. The sixth to eleventh floors, inclusive, are devoted entirely to technical departments of the stock exchange, such as executive offices, institute rooms, governing board room, etc.

Due to the fact that the building is constructed on an interior lot adjacent to one of the noisiest and busiest streets in Los Angeles, a paramount requirement was that the building be entirely sealed with locked windows, the rooms acoustically treated and thoroughly air conditioned in order to maintain practically constant wet and dry bulb temperatures throughout the year.

Exposures and Nature of Occupancy Make Zoning Necessary

The problem of air conditioning this building presented many difficulties, as the various exposures that certain rooms are subject to and the number of occupants that congregate in certain portions of the building make these various spaces entirely different relative to heating and cooling requirements. The solution of the problem was to separate and divide the building in air condition-

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†Chief engineer with Ralph E. Phillips.

ing system units to take care of the separate exposures and internal conditions. As a result, the entire basement which consists, more or less, of service quarters, is equipped with a supply and exhaust mechanical ventilating system, the supply air being washed only and not conditioned; the basement supply fan also supplies air separately heated and controlled, but not conditioned, to the public spaces on the ground floor.

The main trading room on the second floor, which is 70 ft. wide, 94 ft. long and 40 ft. high, together with the members' lounge and comparison room on the same floor, has a complete air conditioning system consisting of supply and exhaust fans, heating stacks and dehumidifier.

The clearing house on the fifth floor is supplied with two separate air conditioning systems, as one-half of the floor area is exposed to a roof loss and exposed wall, while the other half is exposed only on one wall, with no roof exposure. The structure above, from the sixth floor to eleventh floor inclusive, is supplied from one air conditioning system, but the air ducts are so divided that the conditions of the east wall, west wall and interior room spaces can be separately controlled as to temperature.

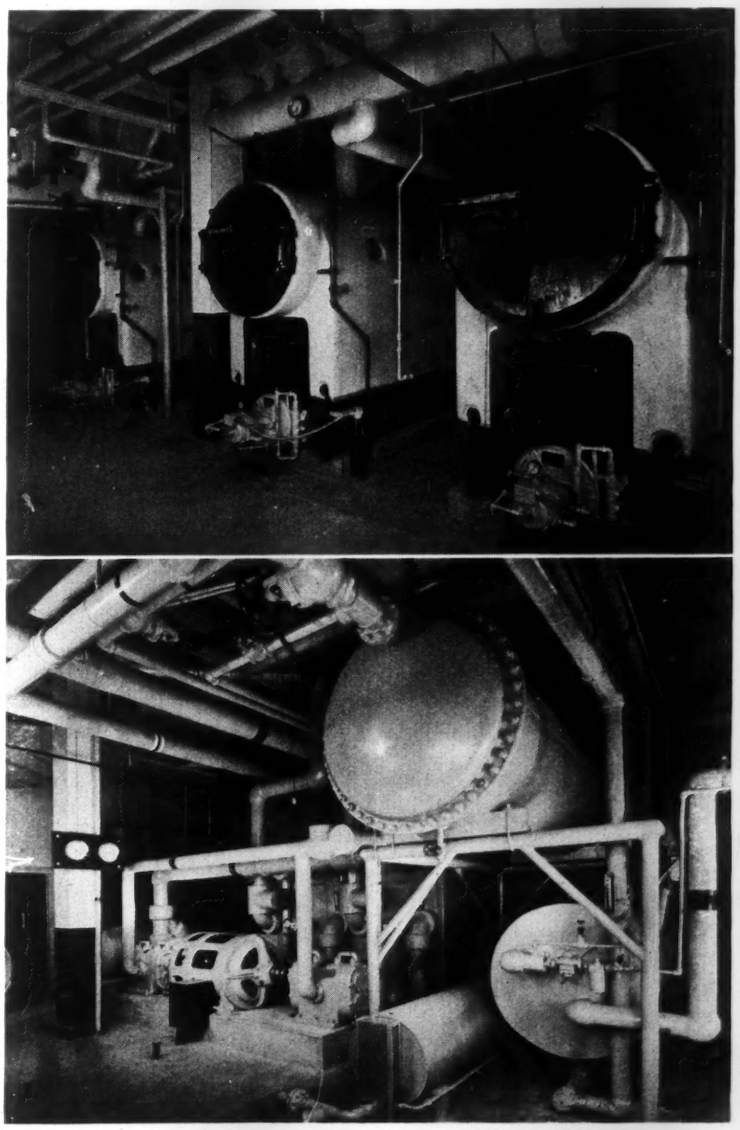
Conditions Maintained

The refrigerating plant was designed to maintain in the air conditioned portions of the building a dry bulb temperature of 75 F and a wet bulb temperature of 64 F when the outside dry bulb is 95 F with a wet bulb of 70 F. Based on these assumptions, a refrigerating plant was required which could deliver 120 tons of refrigeration per 24 hours with a condensing water temperature of 75 F.

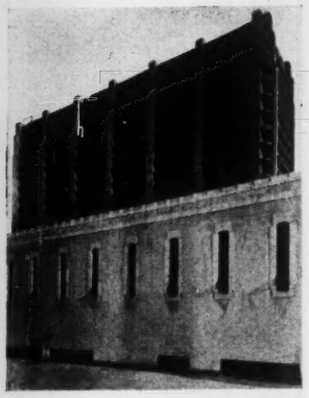
Location of the Equipment

In order to locate the conditioning units as close as possible to the centers of the various loads, the supply and exhaust fans, heating stacks and dehumidifiers for the main trading room and clearing house were located on the third floor level. The equipment for the superstructure, sixth to eleventh floors inclusive, was located in the pent-house and the supply and exhaust fans for the basement were located in the basement machine room. With the dehumidifiers located on two different levels, third floor and pent-house, two closed-type direct expansion water coolers were installed, one water cooler serving the dehumidifiers on the third floor level and the other the dehumidifier at the main roof level.

The pumps for these dehumidifiers are located in the machine room in the basement; they operate on a balanced head, pumping against the pipe friction and spray nozzle pressure only. The refrigerating plant consists of two 60-ton rotary type compressors, using methyl chloride as a refrigerant, direct-driven by one 150-hp. variable speed motor, which is automatically controlled for starting and stopping by an immersion thermostat



TOP — SHOWING THE TWO LOW PRESSURE, OIL-FIRED HEATING BOILERS AND THE SMALL BOILER FOR WATER HEATING IN SUMMER. BOTTOM — REFRIGERATING MACHINERY. WATER COOLERS ARE BEHIND COMPRESSORS AND CONDENSER AND RECEIVERS ARE IN THE FOREGROUND



THE COOLING TOWER

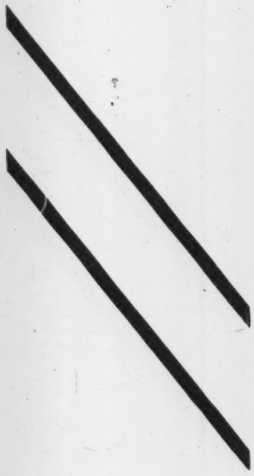


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DIAGRAM OF CHILLED WATER AND CONDENSING WATER PIPING. THE INSERT IS A GENERAL VIEW OF THE MACHINERY ROOM, SHOWING CHILLED WATER AND CONDENSING WATER PUMPS ON THE LEFT, PNEUMATIC TUBE BLOWERS IN THE BACKGROUND, AND REFRIGERATING MACHINERY ON THE RIGHT

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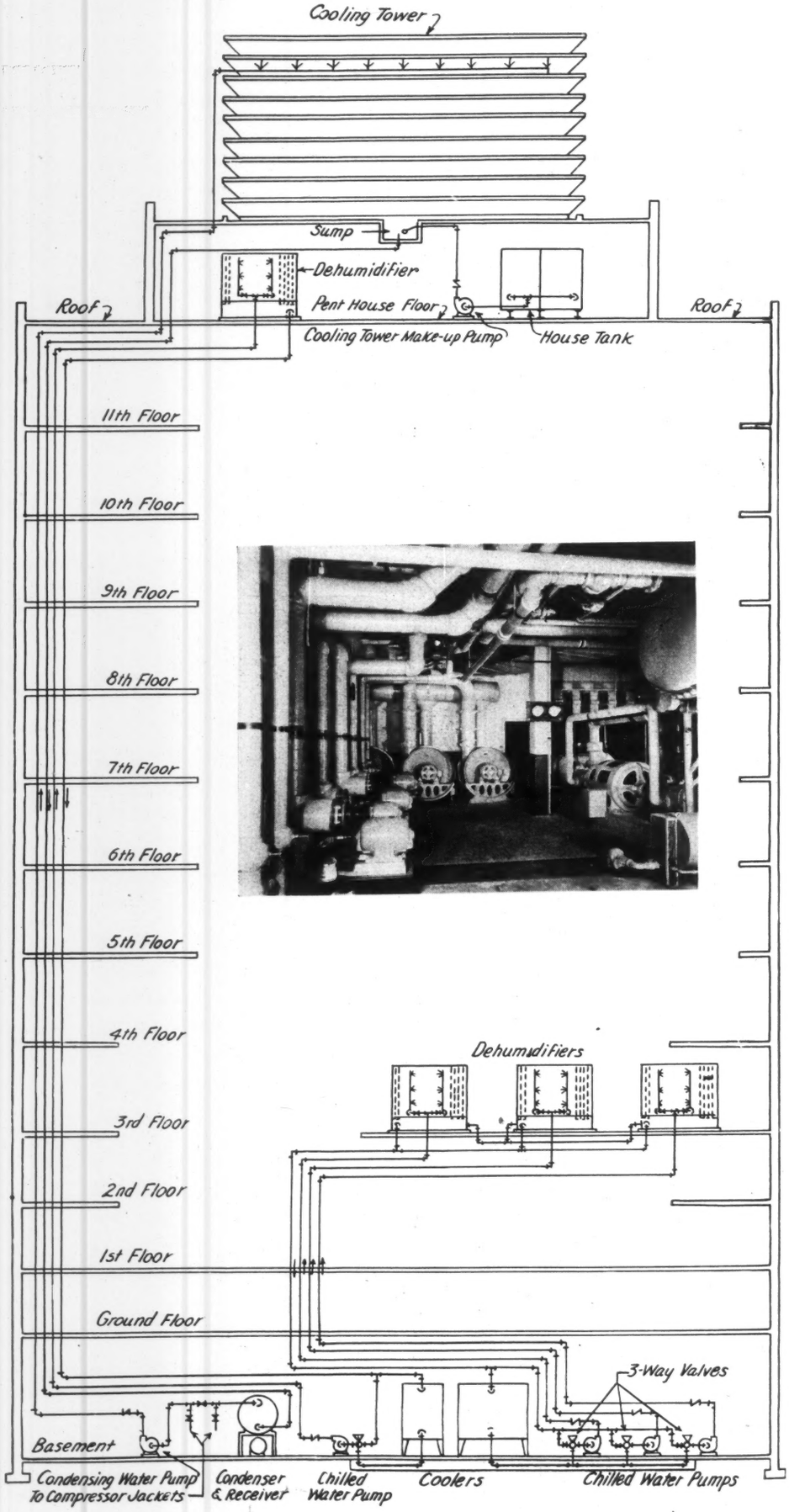


in the water coolers. Condensing water for this plant is cooled in a gravity tower located on the pent-house roof; 480 g.p.m. of water are circulated through this tower by a condenser water pump in the machine room. The cooling tower was architecturally treated in order to form an architectural motif for the building, as it is in full view on the front façade of the structure.

Controlling Conditions and Method of Recirculation

The methods of controlling the air conditions in both summer and winter were carefully considered, as comfortable relative humidity conditions are required throughout the year. The dry bulb temperature during the heating season is controlled by graduated acting thermostatic valves on the heating stacks; the automatic valves are controlled by the duct thermostats and the duct thermostats are in turn controlled by room pilot thermostats. The water temperatures in the dehumidifiers are controlled by the action of three-way valves, which valves automatically supply low temperature chilled water from the coolers to the dehumidifiers or recirculate the water in the dehumidifier or mix the dehumidifier pan water with low temperature water from the coolers to meet the requirements. The three-way valves operate from a wet bulb thermostat located in the return recirculation duct.

All recirculated air passes through the dehumidifier and no recirculation takes place by-passing the dehumidifier. Each air conditioning system is arranged for as high as 80 per cent recirculation, or can be operated by wasting any part of the return air or entirely wasting or exhausting the return. Each supply system, except for the main trading room, also has in conjunction with it an exhaust fan which can either return the air to the suction of the supply fan or partially return it, or entirely waste it.

In the main trading room the recirculated air outlets are located at the floor, the supply at the walls approximately 20 ft. above the floor and the exhaust in the main ceiling exhausting the air through the indirect light reflector boxes; all air exhausted through the light reflector boxes is exhausted by a fan on the third floor level and wasted, but the air returned through the registers located at the floor goes to the suction of the supply fan for recirculation.

Boilers Are Oil-fired

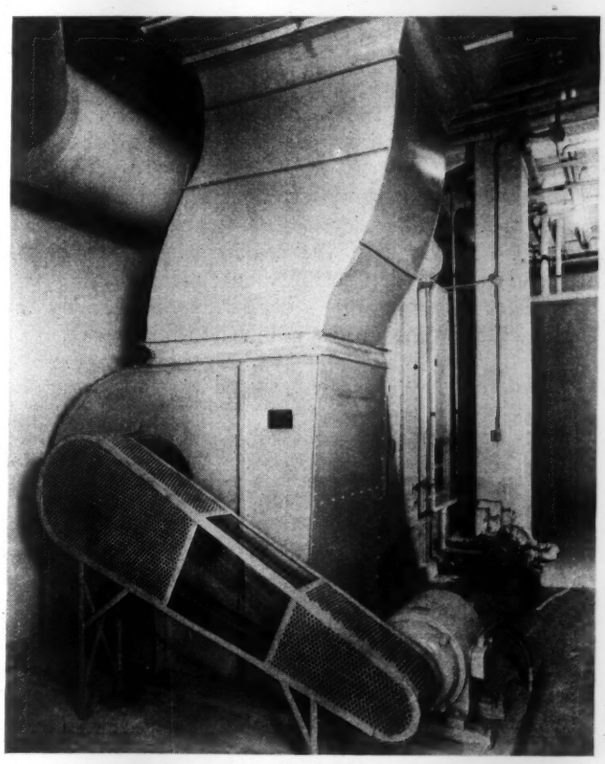
The heating, except for the trading room floor, ground floor and basement, is done entirely by direct radiation supplied from a low pressure steam boiler plant with two oil-fired portable steam boilers and one small portable steam boiler for heating hot water during the summer. The heating system is of two-pipe vacuum-return down-feed design and the supply mains in the roof space are zoned with remote control air-operated valves so that radiation on any exposure of the building may be cut off from the engineer's office. Radiation in such large rooms as the governing board room is thermostatically controlled, and, generally speaking, in all spaces where direct radiation occurs, the ventilating system is only required to introduce the air at approximately room temperature. The main trading room, ground floor and basement are heated by the indirect heating stacks in the ventilating system.

. . . the past few years have proved beyond a doubt that advantages are to be obtained by sealing windows in many types of buildings, acoustically treating the rooms, and maintaining a constant and even temperature at all times . . . the problem of air conditioning in the future rests in the continued development of systems having maximum flexibility for uneven exposures, which are economical in operation, and with an installation cost that will make the air conditioning investment a good investment, considering the increased rentals that tenants are willing to pay for the advantages of air conditioning.

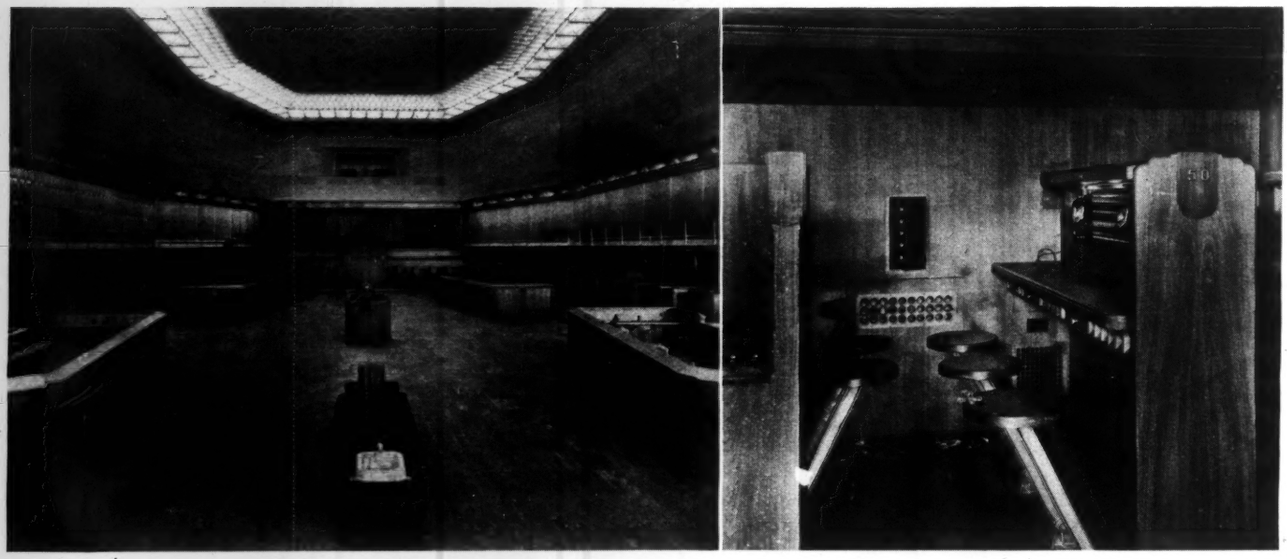
—Ralph E. Phillips

Quantities of Air Supplied

The total quantity of conditioned air supply for the conditioned spaces is 75,200 c.f.m. and as the volume of the conditioned spaces is 460,000 cubic feet, this gives 9.8 air changes throughout the conditioned portions per hour. This air change, of course, is determined entirely by the entering air temperature for which the system is designed and the temperature to be maintained in the rooms conditioned in order to absorb the quantity of heat emitted and transmitted. This air change cannot be



One of the Four Fan Rooms



Left—The Board Room. Air Is Supplied Through a Continuous Bronze Grille Below Bracket Lights Over Quotation Board and Exhausted Through Ceiling Light Box. Recirculated Air Is Returned Through Grilles in Base of the Four Trading Posts and the Members' Booths Around the Wall. Right—One of the 64 Members' Booths, Showing Return Air Grille

considered general for air conditioned structures even in the same city, as the purposes for which the building is to be used and its construction will vary.

In the unconditioned spaces there are 20,000 c. f. m. of air supplied, which changes the air approximately 6.8 times per hour. The exhaust system for the conditioned spaces has the capacity either to recirculate or exhaust 80 per cent of 75,200 c.f.m. or 60,000 c. f. m. The exhaust system for the unconditioned spaces, including the toilet exhaust, has a capacity of 28,600 c. f. m.

Cost of Power

The connected horsepower for the air conditioning apparatus, including the refrigerating plant, fans, and pumps, is 308 hp. and for the unconditioned spaces 32 hp. The electric power for the first year of operation, for the air conditioning systems, the air washing system and the toilet exhaust system, has cost approximately $4,185.00. The logs of the building indicate that the refrigeration is used practically every day of the year, during the hot weather operating nine hours per day and in cold weather one and one-half hours per day.

Cost of Installation

The complete air conditioning, air washing and toilet exhaust systems were installed in the Los Angeles stock exchange building for $72,000.00, which included the complete refrigerating plant and cooling tower, all chilled water and condenser water piping, fans, motors, dehumidifiers, duct work, heating stacks and temperature control. The approximate cost of the washed air supply and exhaust and toilet exhaust system was $10,000.00, giving a net cost for the air conditioning system of $62,000.00.

The new Los Angeles stock exchange building has been in operation for one year, through both cold and hot weather; it has been proved that the air conditioning system has maintained the conditions for which it was designed. During this summer's hot weather, when dry bulb temperatures in Los Angeles were between 95 and 98 F, it was possible to maintain constantly and automatically a wet bulb temperature of 60 F with a dry bulb temperature of 72 F or 50 per cent relative humidity.

NEXT MONTH

SELECTING THE RIGHT SIZE BOILER

Selection of the proper size of steam or hot water heating boiler for a given job frequently is a vexing problem because of the number of factors involved and conflicting practice in rating such boilers. In an article written by Sabin Crocker (which will be published in the February Heating, Piping and Air Conditioning) this question of selection is treated in detail. The tables which will appear with this article assemble in one place the information needed for selecting a heating boiler, and the examples show how they're used. Many engineers will want to preserve this article permanently

NEXT MONTH

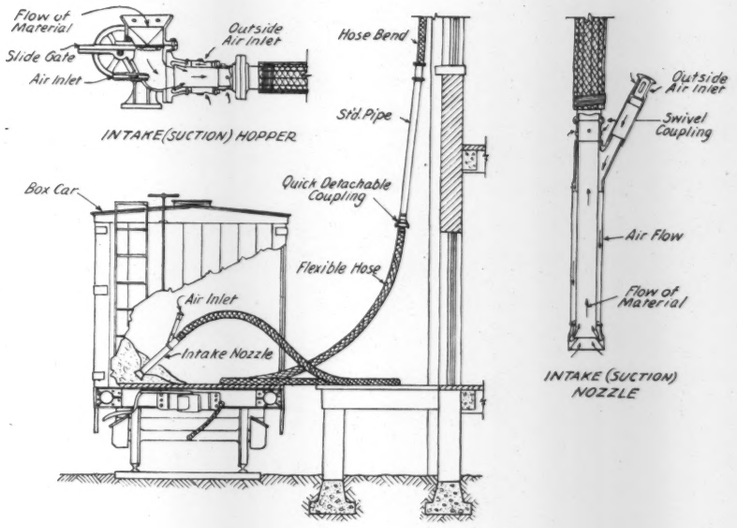


FIG. 1 — TYPICAL INTAKE NOZZLE AND INTAKE HOPPER

By
E. H. deConingh

In many industrial plants, particularly in the chemical industry, unloading and conveying of materials is accomplished economically by means of pneumatic conveying pipe lines. This article tells how piping for this purpose is installed, and discusses such topics as bends for resisting wear, fittings, branches and so forth

Design and Installation of Piping for High Vacuum Pneumatic Conveyor Systems

EARLY pneumatic conveyors operating on the vacuum principle were run under a negative pressure of about 10 in. of mercury, a limit imposed not by design but by the existing development of vacuum producing machinery. Today a 10-in. vacuum system is designated as a medium vacuum system, to distinguish it from the low negative pressures of fan-operated systems, about 1 in. of mercury, on the one hand, and the high vacuum systems, operating at a negative pressure of 18 to 20 in. on the other.

Theoretically, there is no difference between the systems, and the air horsepower to convey a given amount of material a given distance is essentially the same, whether a small quantity of air at high vacuum and velocity is used, or a larger quantity at a correspondingly lower vacuum and velocity. In practice, friction losses from the higher air velocity in the pipe line of a high vacuum system cause the power consumption per pound of material to be somewhat increased.

In Europe, where the pneumatic conveyor was first put to extended practical use, a large amount of grain is handled in ocean-going vessels, and large conveying systems are used to unload this grain from ship to dock warehouse. In this country, however, little grain is imported in ocean-going vessels, and the important application of the suction pneumatic conveyor is found in the unloading of bulk material from railroad car to storage, especially in the chemical industry. For this service, the high vacuum system provides a small diameter pipe line, making possible flexibility of handling at the intake tool. One man can handle an intake nozzle attached to a 3-in. flexible hose with comparative freedom, and can maneuver inside a box-car or a small bin.

The flow of material into the conveying line begins at the intake nozzle or hopper. To unload cars, to draw from storage piles, or to withdraw material from containers, the intake nozzle and a length of flexible hose have been found to be a practical means of initiating the flow. If the material is to be reclaimed from a storage bin, from a pulverizer, or other fixed source, a hopper can be fastened permanently in place, and the pipe line proper can be connected directly to it.

Joining the Piping

For the permanent portion of the pipe line, standard steel piping is used for the majority of installations,

*Technical editor, The Dust Recovering & Conveying Co., Cleveland, Ohio.

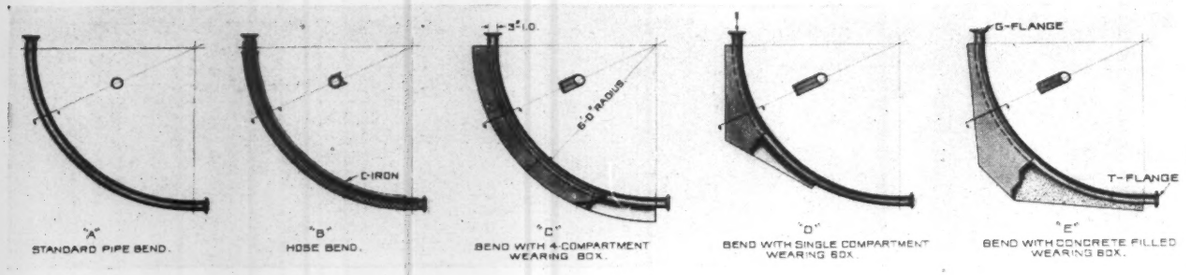


FIG. 2—FORMS OF BENDS USED IN PNEUMATIC CONVEYING LINES

although special considerations sometimes make it necessary to use more expensive material. For example, ingredients for foodstuffs are sometimes conveyed through stainless steel lines; some chemical salts require monel metal construction to withstand corrosion; and intake lines for handling powdered boric acid have been constructed of copper piping.

The inside of the pipe line should present a smooth, unbroken surface to the flow of air and material, and should have as few changes of direction as possible. Perfect alignment of piping is necessary to prevent excessive friction losses, and an offset of an eighth of an inch may cause a serious decrease in conveying capacity. Therefore tongue-and-groove flanges should be used over the full length of line, and care should be taken to thread the flange over the end of the pipe until its face is flush with the pipe cross section, to insure a snug butt joint.

Wear Is Greatest at Bends

Theoretically, the particles of material to be conveyed are evenly distributed in the cross-section of the air stream, but in practice, of course, the larger and heavier particles are concentrated by gravity along the lower portion. The resulting wear on straight lengths of pipe is not serious in conveying most materials, but

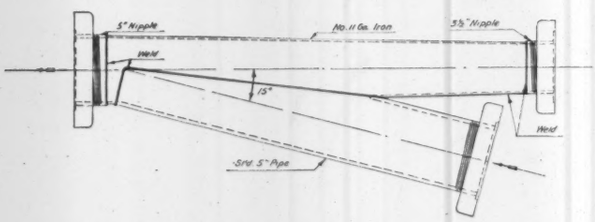


FIG. 3—5-IN. BY 5-IN. BY 3½-IN. Y-FITTING

where there is a steady flow of such abrasive substances as sand or ash, the scouring action makes occasional pipe replacement necessary. Where it is necessary to change the direction of flow, the wear is, of course, greatly increased, since the particles of material are thrown to the outside surface of the curved pipe.

The amount of wear depends upon the velocity of flow through the line, the radius of the bend, and the nature of the material which is being conveyed. Such materials as soda ash, lithopone, and pigments cause little if any abrasive wear on a bend, and if a minimum radius of 6 ft. is maintained, the standard steel pipe bend is satisfactory. There are a number of materials which might be called mildly abrasive, such as lump lime, bauxite, salt cake, etc., which would wear through a steel pipe bend. Rubber hose has been found to be satisfactory for such use, and a tubing of pure rubber of ⅜-in. to ½-in. wall thickness, protected by a helical wire and fabric sheath, stands up very well. The hose can be clamped to a supporting channel iron, and since the wear occurs almost exclusively on the outside of the bend, the hose can be twisted 90 degrees three times before replacement is necessary.

A user of pneumatic conveyors to transport pulverized clinker from furnace ash pit to storage bin, has developed a bend which consists of a steel pipe bend to which has been welded a steel box divided into several compartments. When the wall of the pipe has been worn through, the box fills with ash to form a cushion against further wear. The partitions help to keep the box packed with material, and the life of the bend is extended at the cost of some power loss from eddy currents.

Another type of pipe bend is used on a conveying line through which is maintained a flow of ground phosphate rock, at a rate of 20 tons per hour, 24 hours per day.

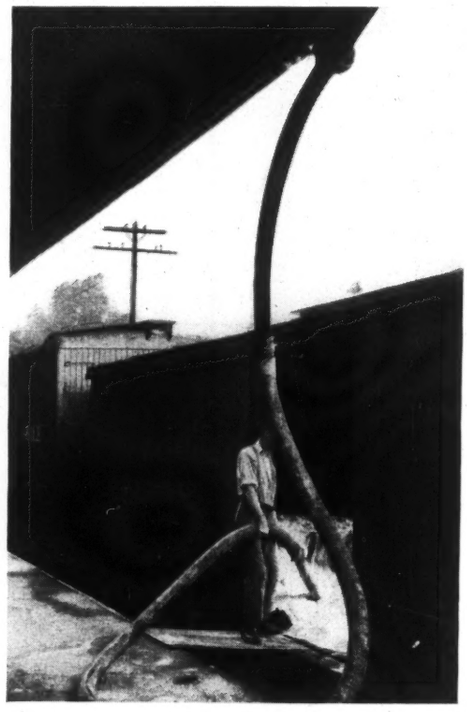


FIG. 4—OPERATOR UNLOADING A CAR OF SODIUM BISULPHATE. MAIN CONVEYING LINE IS SUSPENDED UNDER-EDGE OF CANOPY OVER THE LOADING PLATFORM, WITH SEVERAL DROPS AT CONVENIENT SPACINGS

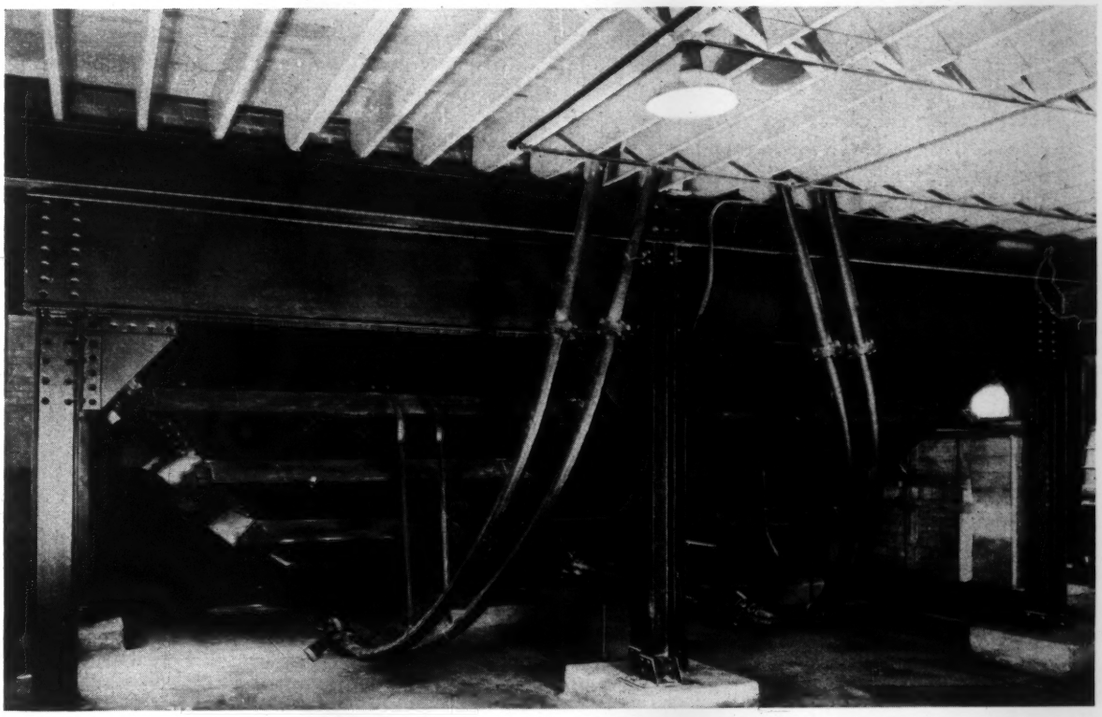


FIG. 5—FOUR INTAKE LINES LEADING FROM HOPPERS UNDER FOUR STORAGE BINS

After the box is welded to the pipe bend, it is filled with concrete through an opening which is subsequently covered with a plate. The abrasive wear hollows out an opening in the pipe wall and a cavity in the concrete, until the enlargement of the cross-section causes deposition of material. This enlargement of the cross-section in the bend causes eddy currents and consequently increased friction loss, but bend replacement has been eliminated.

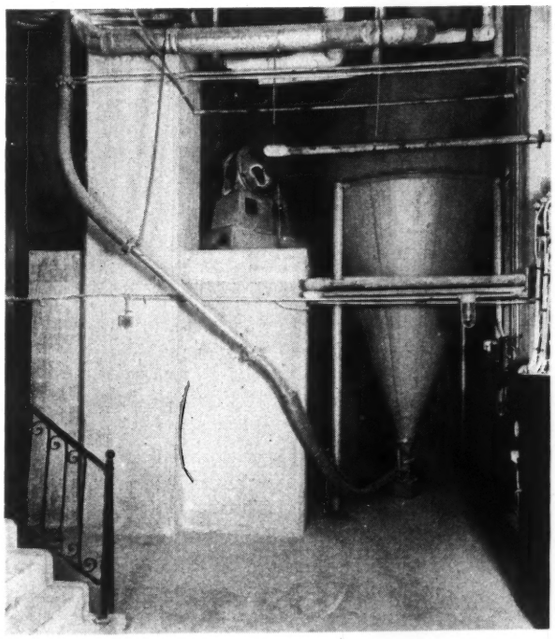


FIG. 6—CONVEYING LINE CONNECTED TO HOPPER UNDER COLLECTING CYCLONE. NOTE THE USE OF RUBBER HOSE FOR COMPOUND BENDS IN CONVEYING LINE

Fittings for Pneumatic Conveying Lines

Branch lines are led into the main line at an angle of not more than 15 degrees, in order to avoid excessive turbulence of the air stream, and cast fittings with machined inside surfaces are generally used. Occasionally an odd sized fitting is required, and to avoid a pattern charge for only one casting, a welded fitting can be used. Fig. 3 shows a fitting used to connect a 5-in. branch to a 3½-in. line, resulting in a 5-in. line. A 5-in. to 3½-in. reducer is first fabricated of ⅛-in. sheet iron, with welded seam. Then a length of 5-in. standard steel pipe is cut off with the torch to make a neat connection at 15 deg. The opening on the reducer is marked to include as much as possible of the longitudinal seam, and is then burned out with the torch. The branch is then welded in place and the threaded nipples are welded to either end of the reducer section.

Where two or more branches of a conveying line are used alternately, valves are necessary to cut out the idle lines. In one type of cut-off valve a plate in which a hole is cut to the same size as the pipe line I.D can be thrown in either of two positions by the hand lever. This plate bears against the valve casting on one side, and on the other is backed up by a sleeve which is held snugly against it by a coil compression spring. Cut-off valves can be operated by solenoids or compressed air pistons and remote control from a central point when they are in inconvenient locations or when other considerations make it advisable.

The conveying pipe line ends with the cast iron fitting which is bolted onto the hopper-bottom cylindrical receiver. The fitting is designed to give the air stream an initial impetus to a spiral path in the receiver, so a replaceable wearing plate is incorporated where abrasive materials are being conveyed.

Conveying Capacity

After the receiving station and fabric filters have been assembled and tested in the shop, it is not desirable to dismantle the equipment for reassembly in the field, because of the difficulty of obtaining as good a job of fitting there as in the factory. For this reason, pneumatic conveyors are usually built of such capacity that the receiving station can be shipped complete on a standard car, and machines to handle as high a capacity as 30 tons per hour have been handled in this way. If a higher capacity is required, it costs little more to use two or three independent systems for ½ or 1/3 the total capacity, and this arrangement offers greater flexibility throughout the whole system.

While a number of these high capacity systems have been built, most specifications call for a capacity of 5 to 10 tons per hour over pipe line distances from 100 to 200 ft. A conveyor of this capacity can unload a box car in an 8-hr. day, through a 3-in. diameter hose and nozzle, and a 3-in. line. This involves a pipe line air velocity of 8,000 to 10,000 ft. per min., which is sufficient to convey most materials, under a negative pressure of 16 to 18 in. of mercury.

The power requirements to convey materials over very long pipe lines usually make such an arrangement impractical, although there are occasional exceptions to the general rule. For example, the Dalecarlia water filtration plant at Washington, D. C., was equipped with a pneumatic conveyor to pick up bauxite, alum, and lime from ground level bins into which the chemicals were dumped from trucks. The trucks had a haul of a little over a mile from the nearest railroad siding. The conveying line was about 110 ft. overall, and the system handled 8 to 10 tons of material per hour from dump bins to storage.

After the system had been in use for some time, it became obvious that the elimination of the trucking would be very desirable and economical. Since the extension of the railroad would have been very costly, the intake pipe line was run to the existing siding, on as direct a line as possible, making a total length of about 1,200 ft. The intake end is now 4 in. in diameter, and is stepped up to a final diameter of 5 in. where the pipe line terminates at the receiving station. The increased friction loss in the longer pipe made it necessary to double the vacuum pump and motor capacity, and the conveying capacity under these conditions has decreased to 4 to 5 tons per hour. This is sufficient to supply the plant, however, and the direct handling from box car to storage bin is a distinct improvement.

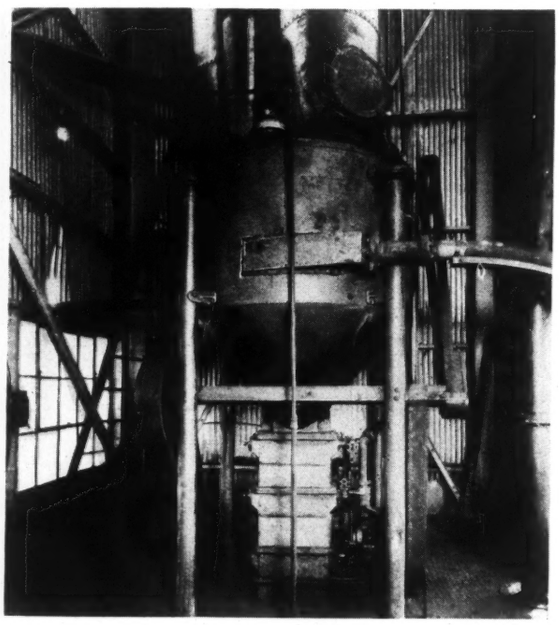


FIG. 7—TERMINUS OF A CONVEYING LINE TO RECLAIM ANNEALING SAND IN A STEEL PLANT. A HOSE BEND, CLAMPED TO A SUPPORTING CHANNEL, IS SHOWN AT THE RIGHT. THE NIPPLE AT THE END OF THE BEND IS CONNECTED TO A REDUCER AND THIS IN TURN IS BOLTED TO THE INTAKE FITTING WITH REMOVABLE WEARING PLATE

Heating a Garage Economically

By Samuel R. Lewis*

A Question

"I WISH to heat a two-car garage by conveying warm air from the boiler room of an apartment building through a duct to the garage. The duct will be about 25 feet long.

"By this means I hope to keep the garage at a temperature above freezing and also make the boiler room 'livable' as it is too hot and it is necessary to keep the windows open.

"What size of duct and fan should I use?"

The Answer

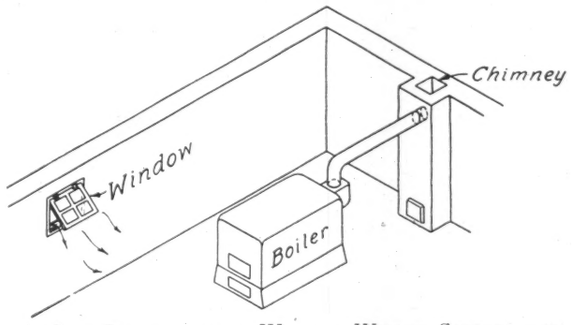
IN THE first place, of course, the boiler room windows *should* be open, because the fuel which heats the building cannot be burned without air. If the boiler room becomes too warm under these conditions it must be that the boiler, the breeching, and the tanks, hot water heater, etc., as well as the pipes, are not insulated as well as they should be. It may be that the entering cool air is not diffused or directed in such manner as to be effective in removing the heat at the ceiling of the boiler room.

In many modern boiler rooms a provision is made for warming the air necessary for combustion by having it absorb the surplus heat at the boiler room ceiling. Usually this is accomplished by extending a duct from the cold air intake parallel with the breeching—sometimes the breeching is placed inside this duct—with the air supply from this duct entering the boiler room close to the ceiling and in such manner as to pass the fireman toward his face rather than toward his back.

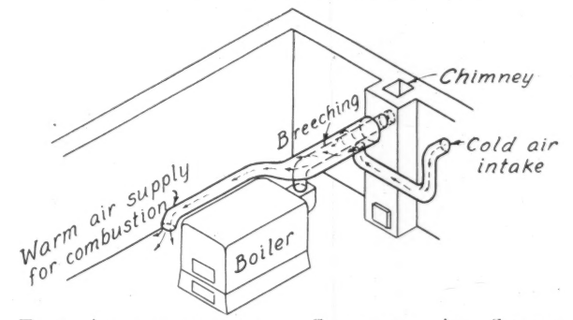
* Consulting engineer, Chicago, Ill.

Return Duct Must Be Used

If the garage is to be heated with the warm air which ordinarily collects at the boiler room ceiling, provision must be made for the return of the partly cooled air from the garage, or a direct air supply from out-of-doors to the boiler room must be arranged to replace the air which is delivered to the garage, in addition to the air required for combustion purposes.



A Cold Draft from a Window Which Strikes the Sometimes-Overheated Fireman on the Back Tends Not Only to Damage the Fireman But Favors the Window Being Nailed Up for the Winter with Consequent Air-Starvation of the Fire



This Arrangement of a Controlled Air Supply Warmed by Otherwise Partly Wasted Heat Is Believed to Be More Efficient and Healthful Than That Shown Above

There is some question whether it is desirable to place the boiler room air in circuit with the air in the garage, since then there will be a tendency for gasoline fumes and exhaust gases to be noticed in the boiler room. The additional duct necessary for the return air from the garage would add considerably to the cost of the installation.

With two ducts, one delivering air from the boiler room to the garage, and one bringing back the partly cooled air there would be no particular danger of interference with the boiler draft by the fan-suction. If there were no return air duct from the garage, however, and if some one should fail to open sufficiently the boiler room window, the fan would be competing with the chimney for air, and might overcome the chimney draft, thus filling the boiler room and the garage with smoke.

It will almost surely be found by experience, if the plant is installed as contemplated, that the surplus heat in the boiler room will be smaller in amount than is anticipated, and that not only the garage but also the boiler room will be too cool in cold weather.

Why Not Use Unit Heater?

The return air duct and the circulating fan will cost very nearly as much, if indeed they do not cost more, than will a unit heating machine comprising a steam convector with its built-in fan.

It will be necessary, in any case, to burn extra fuel in order to warm the garage. If the situation is as stated, it would seem that the most desirable procedure would be to install a small unit heating machine in the garage, provided the steam and return pipes can be extended conveniently. This would save the cost of air ducts between the buildings.

If the grade level relations are not favorable for such a scheme the next choice would be to install a unit heating machine at the boiler room ceiling, having an enclosed supply duct directly from out-of-doors, and delivering its warmed air into the duct which leads to the garage. A damper can be placed in the suction duct to this machine so arranged that as the cold air intake is reduced in size by the damper a direct intake from the boiler room near the ceiling or in it, is increased in size. Thus it would be possible, under supervision, to use the fan to remove surplus heat from the boiler room and to warm the garage, especially in mild weather, at the minimum cost.

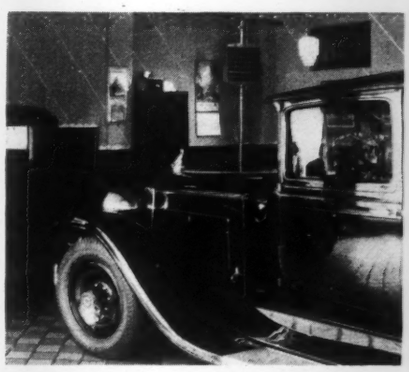
For the ordinary two-car garage a 12-in. diameter tile duct would suffice, and the unit heating machine should be capable of heating about 400 c.f.m. of air from the coldest outside temperature to about 100 F.

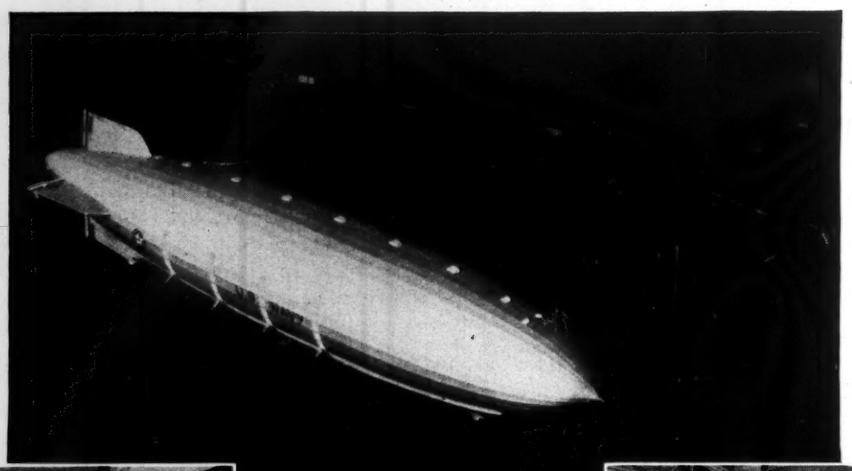
A thermostat in the garage should control the fan in the boiler room, with auxiliary automatic switches at the garage car-entry doors so wired that unless both garage doors are closed, the fan cannot operate.

Or Install Heat Transfer Device

An interesting way to heat a garage is to install a heat transfer device below the boiler water line and alongside the boiler, arranged so that water from the boiler circulates by gravity on one side of the heating surface. On the other side of the heating surface oil or a non-freezing, non-precipitating solution circulates through a separate system of piping to the garage radiators. With such a scheme, even though the garage doors may be left open or even though the garage temperature may drop below freezing, no damage can result.

Unit Heater Installed in an Automobile Showroom

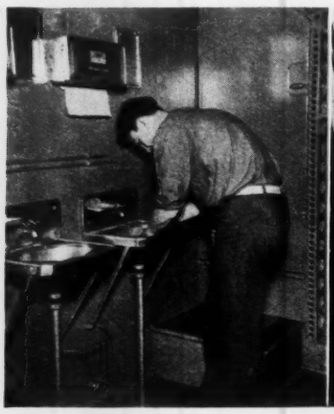


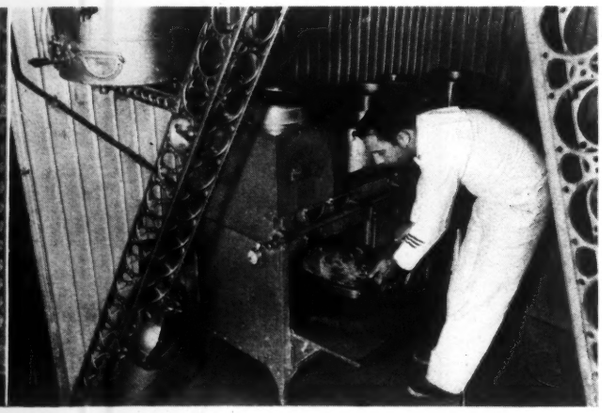


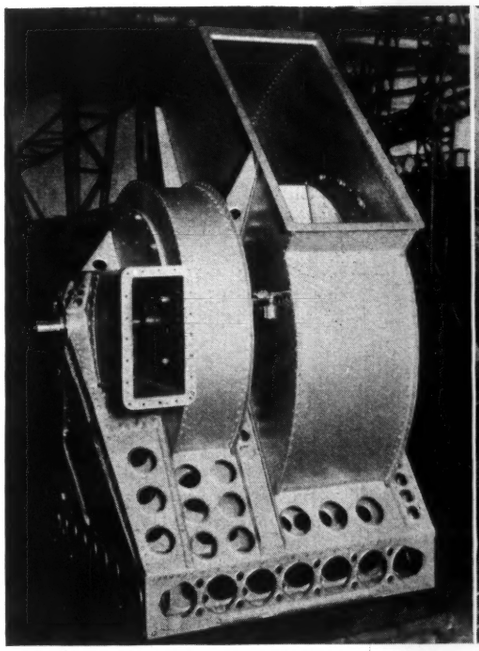
An inspection trip to the Goodyear-Zeppelin hangar at Akron is planned as one of the entertainment features of the annual meeting of the American Society of Heating and Ventilating Engineers this month. These two pages show several views of the airship "Akron" which should be of interest to engineers.

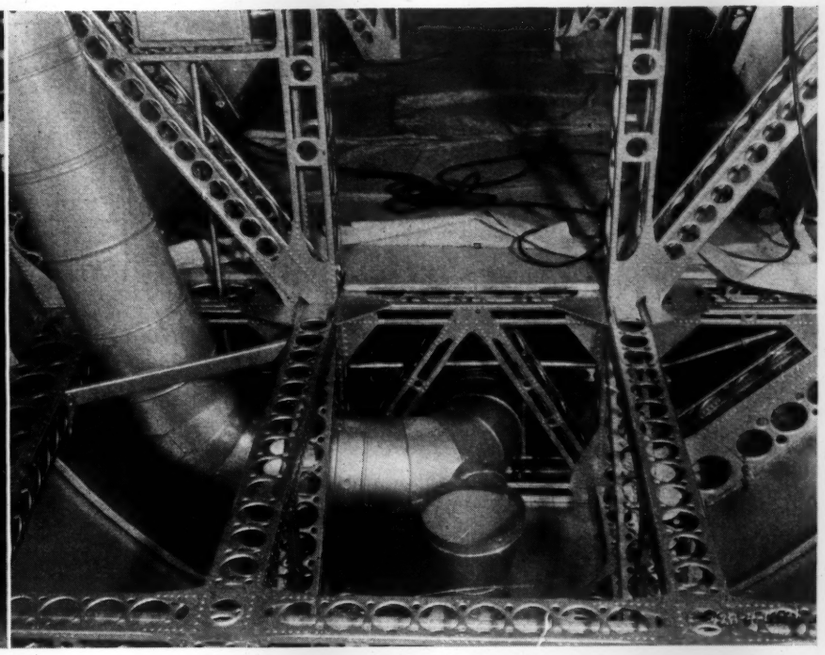
The "Akron" has an over-all length of 785 ft., a maximum diameter of 132.9 ft., and an over-all height of 146.5 ft. Its nominal gas volume is 6,500,000 cu. ft. Its design contains numerous novel features.

At the left below is shown a view of the washroom facilities. The illustration to the right shows the aluminum stove, which weighs 110 pounds, and is large enough to prepare hot meals for the entire crew of 53 men and 10 officers.

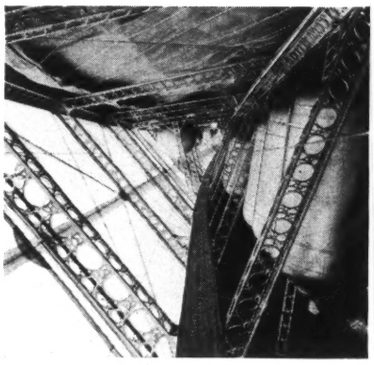




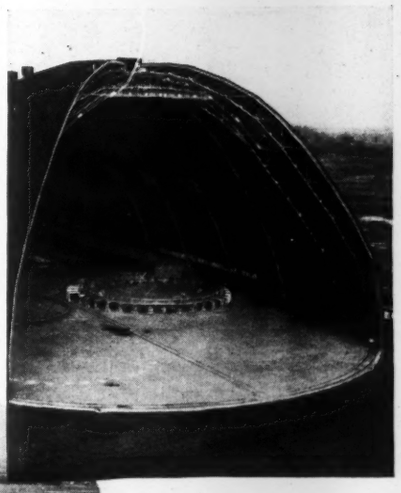


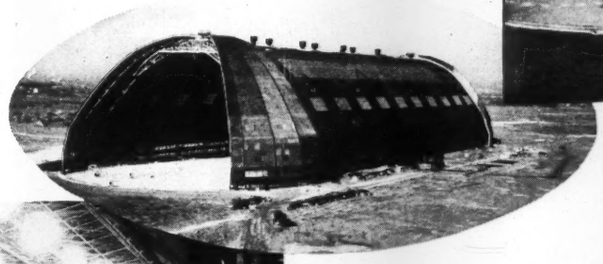


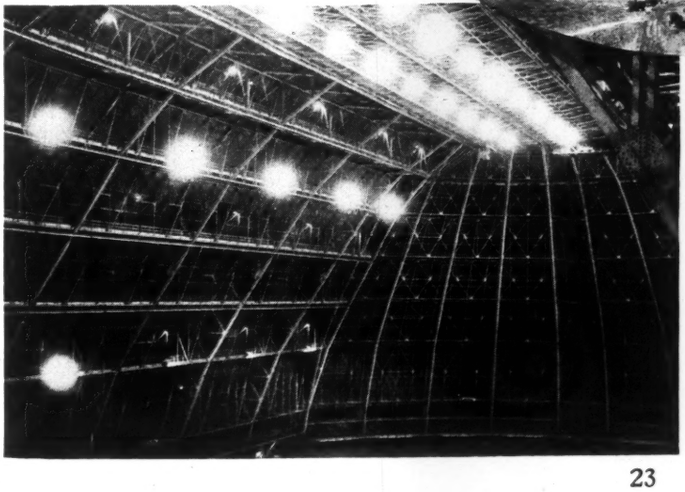
At the top of this page, to the left, is illustrated a ventilating fan for the "Akron." The warm air ducts for heating are shown in the picture at the right.



The fuel oil tanks are illustrated at the left. Provisions for the storage of 62 tons of gasoline and 54 tons of ballast water are made. To compensate for the weight of gasoline burned, a water recovery condenser system is provided for each motor.







The world's largest airship factory and dock is illustrated in these three views. The "Akron" was constructed in it, and in the structure the "ZRS-5," sister ship of the "Akron," is being built. The "Akron" is based at Lakehurst, N. J. Offices and shops are heated by gas-steam radiators, with electric heaters used on the working platforms in cold weather. The quantity of air displaced by the ship had to be considered when determining the number and size of the natural ventilators.

In this article, the author gives information on the various factors that must be considered in selecting vacuum pumps and shows how to apply this information by means of typical problems. The first example deals with a machine which uses a vacuum to handle sheets of paper; another example concerns a vacuum pump in a power plant.

Solving Vacuum Pump Problems

By S. B. Redfield*

THERE are numerous interesting problems regarding the performance and testing of vacuum pumps which may be solved by the information contained in a previous article by the writer.[1] For the purposes of illustration it will be worth while to work out a problem or two by the methods there suggested. The problems in this present article, while thoroughly practical, are purely hypothetical and do not refer to any actual installation. The conditions and figures have been chosen with the deliberate intent of bringing out as many important points as possible.

Selecting the Proper Size of Pump

For the first problem, assume that there is a certain machine used in book binding which uses a vacuum to handle sheets of paper. It is desired to furnish with each machine a vacuum pump of suitable size to maintain a 24-in. vacuum with a 30-in. barometer. How large a pump must be furnished?

Test Run; Results

As a preliminary step, a test was run on an existing paper handling machine of a smaller size and it was found that a certain vacuum pump with a 4½ x 2 single-acting cylinder running at 700 r.p.m. would produce a vacuum of 12.7 in. of mercury. The barometer stood at 29.3 in. during the test and the temperature in the pump suction pipe was 70 F. It is known that the new machine will have 25 per cent more sections than the old machine and so will have 25 per cent more suction nozzles. Also, as stated in the problem, the desired vacuum is 24 in., on a 30-in. barometer; whereas the test pump on the small machine pulled only 12.7 in. vacuum with a barometer of 29.3 in.

Approximate Volumetric Efficiency Line

Referring to the former article[1] on vacuum pump testing it will be seen that an approximate volumetric efficiency line may be drawn by first observing the *dead end* or *closed intake* vacuum that the pump will pull. Under this condition it was shown that the volumetric efficiency of the pump cylinder is zero. It can also be assumed that with inlet and outlet of the pump wide open, and pumping no vacuum at all, the volumetric efficiency of the pump may be from, say, 85 per cent up to 96 or 97 per cent. The actual figure depends very much upon machine size, valve construction, clearance,

* Engineer, Compressor Department, Fuller Company, Catasauqua, Pa.
[1] "How to Make Nozzle Tests of Vacuum Pumps," by S. B. Redfield. Page 844, HEATING, PIPING AND AIR CONDITIONING, October, 1930.

piston leakage, etc. For pumps of 150 ft. displacement or more, this figure may easily be from 93 to 96 per cent. For larger machines it may be as high as 98 per cent. For the very small units often used on such paper handling devices as we are now considering, 90 per cent volumetric efficiency at zero vacuum will be considered good enough for an estimate. Most work is much higher up on the vacuum scale, where this zero-vacuum volume will not have a critical influence.

It was shown in the former article that if a straight line is drawn between the points of dead end, or maximum vacuum with closed intake (zero volumetric efficiency) and the point of maximum volumetric efficiency with zero vacuum, this line will give a fair estimate of the volume performance of the pump. This line will closely represent the volumetric efficiency of the pump if the quantity of air handled is expressed as *free air*, or cubic feet at atmospheric pressure and at the temperature of the pump cylinder intake.

Assume that after testing the first paper handling machine, the pump intake right at the cylinder casting entrance was closed and it was found that the dead end vacuum was 27.8 in., the barometer being still at 29.3 in. Further, assume that with no vacuum at all on the pump suction the volumetric efficiency of this little pump was taken as 90 per cent. The diagonal line can now be drawn, and as in the previous article, a little "sag" will be put into the line near its high vacuum end. Fig. 1 shows such a line.

It is often customary to plot vacuum as per cent of the barometer so in this diagram the horizontal distances represent vacuum in per cent of the barometer while vertical distances represent volumetric efficiencies. It is repeated that these volumetric efficiency values are obtained by expressing the output of the pump in cubic feet of air per minute measured at atmospheric pressure and at the temperature of the pump suction. This volume of atmospheric air divided by the displacement of the pump cylinder, in cubic feet per minute, is the **atmospheric volumetric efficiency.**

The diagonal line in Fig. 1 is a fair representation of the variation of this atmospheric volumetric efficiency in a small pump such as is being considered. The dead end vacuum is 27.8/29.3 or 95 per cent of the barometer, as determined by closed intake test. The true volumetric efficiency line, showing the air volume handled expressed at the pressure and temperature conditions of the pump suction is another curved line, as fully set forth in the previous article. In this problem, the atmospheric volumetric efficiency line only will be used, so the other one is not drawn in.

Effect of Vacuum on Amount of Air Flowing

In the previous article a curious phenomenon was also discussed. This is the fact that, as long as the absolute pressure of the vacuum into which air is flowing is less than half the upstream absolute pressure (which in this case is atmospheric) a further increase in vacuum (reduction in downstream absolute pressure) will not increase the quantity of air flowing. Thus, the quantity of air flowing from atmosphere into a 28-in. vacuum is no more nor less than will flow through the same opening from atmosphere into a 15-in. vacuum. In both cases the downstream absolute pressure is less than half that on the upstream side. As opposed to this, if the downstream absolute pressure is more than half the upstream pressure, the actual value of the vacuum into which the air is flowing will influence the quantity of flow. Thus, flowing from atmosphere into a vacuum of 14 in., more air will pass than if flowing from atmosphere into an 8-in. vacuum. In these two latter

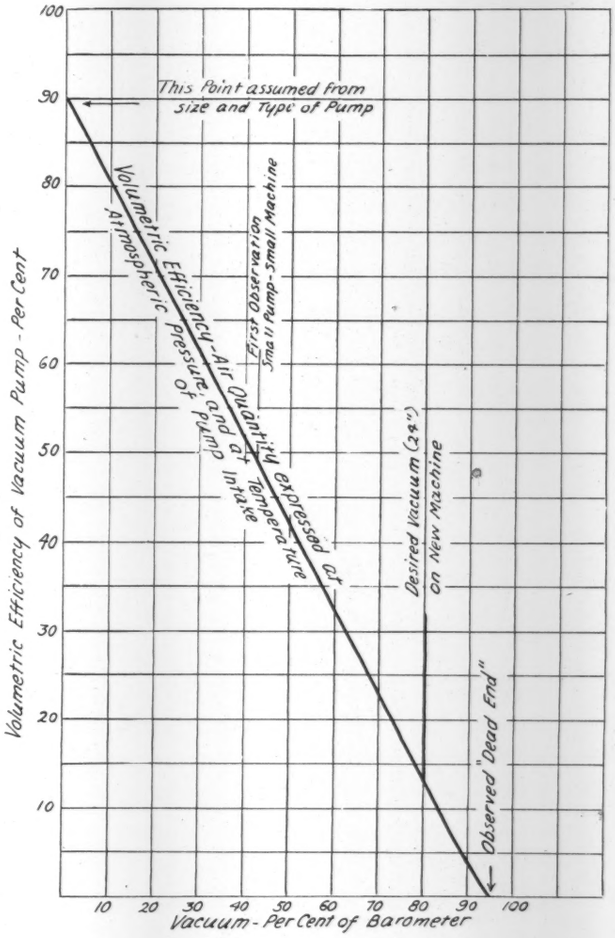

This Point assumed from Size and Type of Pump
Volumetric Efficiency - Air Quantity expressed at Atmospheric Pressure and at Temperature of Pump Intake
First Observation Small Pump - Small Machine
Desired Vacuum (24") on New Machine
Observed "Dead End"
Volumetric Efficiency or Vacuum Pump - Per Cent
Vacuum - Per Cent of Barometer
100
90
80
70
60
50
40
30
20
10
10 20 30 40 50 60 70 80 90 100

FIG. 1—CHARACTERISTIC VOLUMETRIC EFFICIENCY CURVE OF A SMALL VACUUM PUMP

cases the downstream pressure is more than half the upstream pressure (both absolute). In all these examples the air "quantity" may be expressed as cubic feet at atmospheric pressure and temperature, or as weight of air in pounds per second or per minute. It is not expressed at downstream conditions because these are different in each example.

Formulas for Flow

It has been said that the dividing line or critical pressure is half the upstream pressure (always absolute); the actual critical downstream pressure is 0.53 of the upstream absolute pressure. The value "one-half" is a figure easy to remember. Due to this curious change in flow, it is necessary to figure the quantity of flow by two different formulas. Two well known formulas developed by Fleigner many years ago, are as follows (all pressures are absolute):

When the downstream absolute pressure is less than the critical, ($P_2 < 0.53\ P_1$):

$$G = 0.53\ F\ \frac{P_1}{\sqrt{T_1}}$$

When the downstream absolute pressure is greater than the critical (P_2 0.53 P_1):

$$G = 1.06\ F \sqrt{\frac{P_2\ (P_1 - P_2)}{T_1}}$$

Where
G = flow of air, lb. per sec.
F = area of orifice, sq. in.
P_1 = upstream absolute pressure, lb. per sq. in.
P_2 = downstream absolute pressure, lb. per sq. in.
T_1 = absolute temperature of flowing air (upstream side)

These formulas assume an orifice having a well-rounded entrance. If the entrance edges are sharp, it is necessary to apply a flow coefficient which may be as low as 0.56. In this problem the coefficient can be neglected, assuming it the same in both cases, as the actual areas do not affect the problem.

In this problem the barometer was said to be 29.3 inches, and the small pump pulled 12.7-in. vacuum when attached to the first paper handling machine. Pressures may be expressed either as inches of mercury direct, or they may be changed to pounds per square inch. At ordinary atmospheric temperature, pressure per square inch is 0.491 times the mercury column, or 10 in. of mercury column is equivalent to 4.91 lb. per sq. in. To visualize relative values of vacuums and absolute pressures, lines may be drawn as in Fig. 2. At the left end of the lines are shown the absolute pressures in pounds per square inch and at the right the same absolute pressures in inches of mercury. The top line is the atmospheric pressure and the bottom line is absolute zero pressure. The distances between lines, *measured downward*, represent vacuum gage readings. The barometer has been assumed as 29.3 in. as in the problem.

Pump Displacement

In the test of the paper handler a vacuum of 12.7 in. was observed. This is 12.7/29.3 or a 43.4 per cent vacuum. From Fig. 1 it may be seen that the probable volumetric efficiency (atmospheric air) of the pump was around 48 per cent. This point is marked in Fig. 1. The displacement of a 4½ x 2 single acting vacuum pump is the area of the piston multiplied by the stroke and by the r.p.m. The area of a 4½-in. piston is 15.9 sq. in., so displacement is

$$\frac{15.9 \times 2 \times 700}{1728} = 12.9 \text{ c. f. m.}$$

With an approximate volumetric efficiency (from Fig. 1) of 48 per cent, this pump was handling about

$$0.48 \times 12.9 = 6.19 \text{ c. f. m. of air}$$

measured at atmospheric pressure (29.3 barometer, 14.4 lb. per sq. in. absolute) and at the temperature of the pump intake, 70 F, or 530 F absolute. We now want to know, by the problem, how much pump displacement to provide for a paper handling machine with 25 per cent more nozzles and to produce a 24-in. vacuum with a 30-in. barometer.

Determining Increased Quantity of Air

The mathematical way to determine the increased quantity of air would be to make a ratio using the values

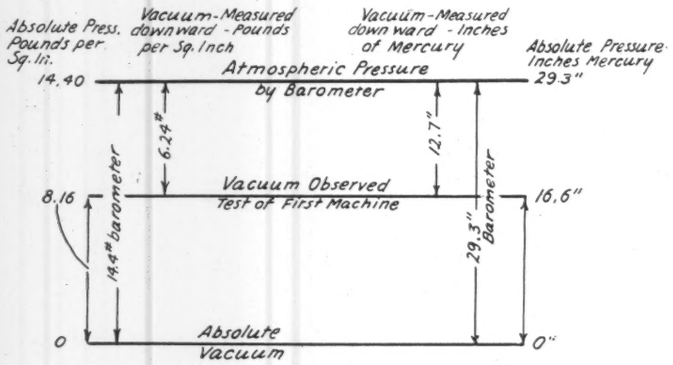


FIG. 2—TEST CONDITIONS IN THE PROBLEM

in the problem in the two formulas above given. This is not difficult for those familiar with such calculations; however, the method of "equivalent areas" is simpler. This latter method will be used first, but afterwards the ratio will be expressed directly by inserting the values in the two formulas and dividing one by the other. This ratio will be the ratio of the air quantities under the two conditions.

It should be observed by studying Fig. 2 that under the test conditions, when a 12.7-in. vacuum was drawn, the absolute pressure on the vacuum side was 29.3 — 12.7 = 16.6 in. of mercury, or 8.16 lb. per sq. in. This pressure is greater than half the observed atmospheric pressure of 29.3 in. of mercury, or 14.40 lb. per sq. in. Therefore, to calculate the flow, the second of the two formulas would be used. For the case where a "vacuum of 24" is desired, the barometer being 30 in., the absolute pressure of the vacuum is 30 — 24 = 6 in. of mercury, or 2.95 lb. per sq. in. This is less than half the atmospheric pressure of 30 in., or 14.73 lb. per sq. in. For this condition the first of the two formulas must be used.

Equivalent Area Method

By the "equivalent area" method, the appropriate flow formula will be used to figure what equivalent nozzle area would produce the flow already determined from the test data (6.19 cubic feet per minute). To this area will be added 25 per cent to provide for the paper machine with 25 per cent more nozzles; then the appropriate formula will be used to figure how much air will flow through this new equivalent nozzle area under the new

condition (24-in. vacuum with a 30-in. barometer).

For the test conditions, using the second formula

$$G = 1.06\ F \sqrt{\frac{P_2\ (P_1 - P_2)}{T_1}}$$

so

$$F = \frac{G}{1.06 \sqrt{\frac{P_2\ (P_1 - P_2)}{T_1}}}$$

G is the weight of air per second corresponding to the observed flow of 6.19 c.f.m. at atmospheric pressure and 70 F. Air at 14.7 lb. absolute pressure and 70 F weighs 0.075 lb. per cu. ft. The atmospheric pressure in this first case is 14.4 lb .per sq. in. (29.3 barometer). Therefore the atmospheric air in the room must weigh

$$0.075 \times \frac{14.4}{14.7} = .0735 \text{ lb. per cu. ft.}$$

The next problem is to figure how much air will pass through this new nozzle area with a 24-in. vacuum and a 30-in. barometer. As already explained the coefficient of flow is neglected, assuming it to be the same in both cases; or assuming a well rounded entrance to each nozzle or orifice. For this case Fig. 3 gives a diagram of the pressure conditions. It shows that for 24-in. vacuum the downstream absolute pressure is 6 in. of mercury, or 2.95 lb. per sq. in. This is much less than half the upstream (atmospheric) absolute pressure (30 in. of mercury or 14.73 lb.) Therefore, to calculate the flow the first formula, for downstream pressure less than critical, must be used. Therefore:

$F = .02885$ sq. in.
$P_1 = 14.73$
$T_1 = 530$ abs. as before.

Therefore,

$$G = \frac{0.53 \times 0.02885 \times 14.73}{\sqrt{530}} = .00978 \text{ lb. per sec.}$$

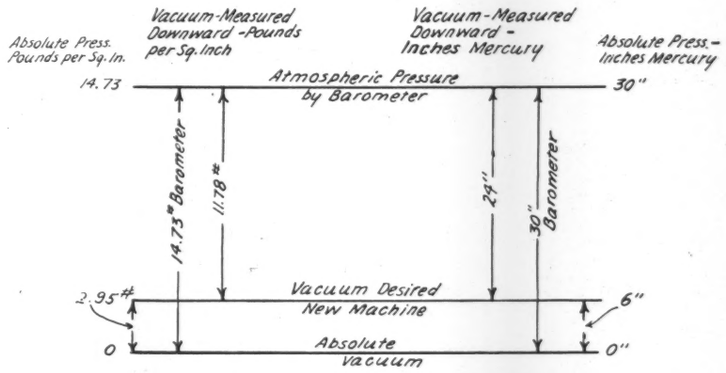


FIG. 3—CONDITIONS DESIRED IN NEW MACHINE (See problem in text)

Therefore,

$$G = \frac{6.19 \times .0735}{60} = .00758 \text{ lb per sec.}$$

Also (See Fig. 2)

$P_2 = 8.16$; $P_1 = 14.4$; $(P_1 - P_2) = 14.4\text{-}8.16 = 6.24$
$T_1 = 70 + 460 = 530$

So the equivalent area

$$F = \frac{.00758}{1.06 \sqrt{\frac{8.16 \times 6.24}{530}}} = .0231 \text{ sq. in.}$$

Of course, the actual total area of the vacuum nozzles would probably be much larger because much of the vacuum at the pump is produced by the air friction through the piping system. For the sake of simplicity it is assumed the vacuum is measured at the pump intake connection in each case. Also, in a case like this, the air flow would be intermittent as the paper sheets were alternately picked up and dropped. This would cause wide fluctuations in vacuum near the nozzles; but close to the pump these variations would doubtless be smoothed out to an average value. It is this average value which is considered in this problem.

Flow of Air Through New Nozzle Area

The above result is an "equivalent area" of nozzle in the first machine under test; it was stated that the new machine will have 25 per cent more nozzles. Therefore the equivalent area of the nozzles in the new machine would be $1.25 \times .0231 = .02885$ sq. in.

As all the measurements for cubic feet of air are expressed at atmospheric pressure and temperature of pump intake (70 F), the cubic feet of atmospheric air corresponding to the above weight can be obtained by dividing by the weight of the atmospheric air in pounds per cubic foot. Again, air at 14.7 lb. per sq. in. absolute pressure and 70 F weighs 0.075 lb. per cu. ft., so air at 14.73 lb. per sq. in. (30-in. barometer) and 70 F will weigh

$$0.075 \times \frac{14.73}{14.7} = .0751 \text{ lb. per cu. ft.}$$

The flow under the new conditions in the new paper handling machine will then be

$$\frac{0.00978}{0.0751} \times 60 = 7.81 \text{ c. f. m.}$$

measured at atmospheric pressure (14.73 lb. per sq. in.) and 70 F.

Selecting the Pump

The next step is to find out what size vacuum pump to select. Before doing this the dead end point and atmospheric volumetric efficiency of the pump must be known. This information can be furnished by the builders but for this problem, the curve in Fig. 1 can be assumed to do for this pump as well as the original one. This curve shows an atmospheric volumetric efficiency of 13 per cent at 80 per cent vacuum, which is 24 in. on a

30-in. barometer, as specified. Therefore to deliver 7.81 cu. ft. the pump displacement should be

$$PD = \frac{7.81}{13} \times 100 = 60.1 \text{ c. f. m.}$$

An 8 × 5 single acting vacuum pump has a displacement of 65.5 c.f.m. at 450 r.p.m. It could run therefore at

$$\frac{60.1}{65.5} \times 450 = 414 \text{ r. p. m.}$$

However, it would be good policy to allow it to run at its rated speed of 450 r.p.m. to allow for any extra flow which might occur due to leakage into the system.

Solving with "Flow Ratio"

If it is desired to solve this problem without resorting to the "equivalent areas," the ratio of the new flow quantity to the test quantity will be the ratios of the two formulas, each one containing the correct pressures, etc., corresponding to the two conditions and considering the value of F in the second case to be $1.25 \times F$ in the first case to allow for the greater number of nozzles in the new machine.

Then G represents the flow in the test machine and G_1 the flow in the new machine, so

$$\frac{\text{Flow in new machine}}{\text{Flow in test machine}} = \frac{G_1}{G} = \frac{\dfrac{0.53 \times 1.25\,F \times 14.73}{\sqrt{530}}}{1.06\,F\sqrt{\dfrac{8.16 \times 6.24}{530}}}$$

$$\frac{G_1}{G} = \frac{0.53 \times 1.25 \times 14.73}{1.06 \times \sqrt{8.16 \times 6.24}}$$

$$= \frac{0.53 \times 1.25 \times 14.73}{1.06 \times 7.13} = 1.292$$

Then $G_1 = 1.292\ G$

From the displacement of the first pump the observed vacuum and its corresponding volumetric efficiency, G was found to be .00758 lb. of air per sec. Therefore $G_1 = 1.292 \times .00758 = .00978$ lb. per sec.

This is the same flow weight figured before, with a 24-in. vacuum with a 30-in. barometer and an increase of 25 per cent in nozzle area. Again, it corresponds to

$$\frac{0.00978 \times 60}{0.0751} = 7.81 \text{ cu. ft. at 14.73 lb per sq. in. and 70 deg F,}$$

to be handled per minute by the larger pump on the new machine, as previously calculated.

It should be noted that the ratios of the two flows were taken in pounds and not in cubic feet. This is because in the first case the barometer was 29.3 in. and in the second it was 30 in. This makes the air in the first case lighter per cubic foot than in the second case, so the ratio of the flows in cubic feet measured at the two different atmospheric pressures is not the same as the ratio of the flows in pounds.

Another point to note is that the ratio of the flows is as 1.292 is to 1; whereas, the nozzle areas are to each other as 1.25 is to 1. With only 12.7 in. vacuum in the first case and 24 in. vacuum in the second case, it would seem as if there should be a greater difference. However, this is because 12.7 in. vacuum of the first case is not far from the critical pressure. The flow will increase as the vacuum increases only to the point when the downstream absolute pressure is 0.53 of the absolute upstream pressure. The critical ratio would occur when the downstream absolute pressure is

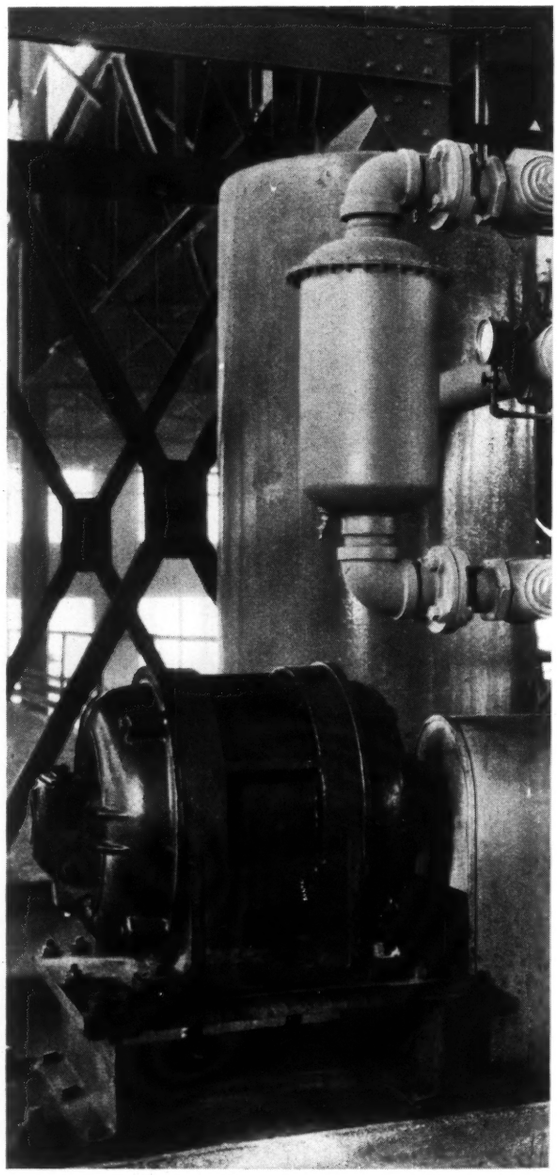
$$0.53 \times 29.3 = 15.53 \text{ in. of mercury.}$$

The vacuum would then be $29.3 - 15.53 = 13.77$ in. As the test vacuum was 12.7 it is easily seen that the flow could increase only to the point when the vacuum would become 13.77 in. or only about 1 in. more than was observed. After this critical ratio the increase of vacuum to 24 in. had no effect upon the flow.

In spite of the fact that the flow in the second case was only 1.292 times that in the first place, it is nevertheless a fact that the pump displacement required in the second case was

$$\frac{60.1}{12.9} \text{ or } 4.66 \text{ times as much as in the first case.}$$

This takes into account not only the increased flow due to increased vacuum and increased number of nozzles, but also the much decreased volumetric efficiency of the



pump when working at 24 in. vacuum instead of at 12.7 in. vacuum. This comparison brings out the importance of each one of these factors.

Air Leakage into a System

In a certain power plant a "dry" vacuum pump having a piston displacement of 1085 c.f.m. attached to a condensing system produces a vacuum of 26.70 in. At the same time the temperature of the pump intake, handling air and steam vapor, is observed to be 109 F. The barometer is 29.60 in. How much air is leaking into the system?

The absolute pressure at the pump intake is 29.60 — 26.70 = 2.90 in. of mercury. Saturated steam at this pressure has a temperature of 114 F. The presence of the air thus makes it necessary to carry a temperature 5 degrees lower than the vacuum would require if all steam; or contrariwise, the vacuum is not as high as the temperature of 109 F ought to produce in pure steam. (A hot well temperature of 109 degrees corresponds to an absolute steam pressure of 2.515 in. of mercury or 29.60 — 2.515 = 27.08 in. vacuum on the observed barometer, instead of 26.7 in).

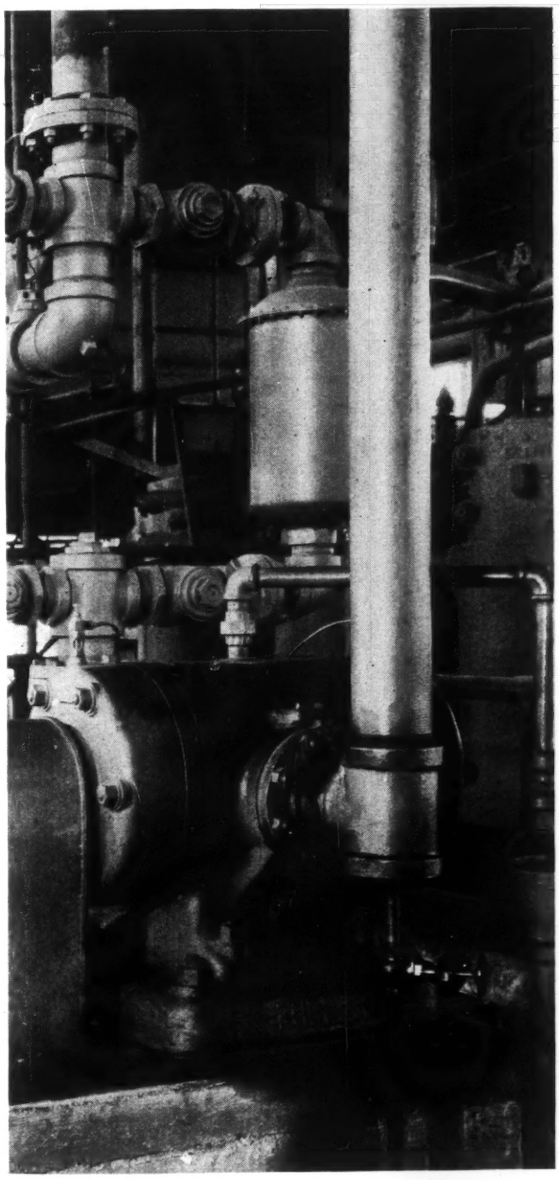


FIG. 4 — ROTARY TYPE VACUUM PUMP IN AN ASBESTOS PLANT. THE SYSTEM HANDLES ASBESTOS SHEETS IN THE MANNER DESCRIBED IN THIS ARTICLE

Assume that the atmospheric volumetric efficiency line has been drawn as previously described, from a test of the pump showing a "dead end" vacuum of 96.6 per cent of the barometer. This would be 29 in. on a 30-in. barometer. Such a line would indicate an approximate *atmospheric volumetric efficiency* of about 5.4 per cent at the observed vacuum of 26.70 in. on a 29.60 in. barometer, or 90.2 per cent vacuum.

The volume of steam and air mixture therefore being handled by the pump is

$$0.054 \times 1085 = 58.6 \text{ c. f. m.}$$

expressed at atmospheric pressure of 29.6 in. barometer. It is also at a temperature of 109 F. This mixture is going into the pump cylinder at the absolute pressure of the observed vacuum, or 2.9 in. of mercury and therefore the real volume of mixture handled is $58.6 \times 29.6/2.9 = 598$ cubic feet expressed at the conditions of vacuum and temperature observed at the pump intake.

The law of mixtures of gases and vapors states that each constituent occupies the entire space and each exerts its own partial pressure, the total observed pressure being the sum of the two partial pressures. The mixture considered is steam and air at a temperature of 109 F. The steam tables show that the absolute pressure of saturated steam at 109 F is 2.515 in. of mercury. The vacuum gage shows that the absolute pressure at the pump intake is 2.9 in. of mercury. This is the total pressure made up of the sum of the partial pressures of the air and of the steam. Therefore, the partial pressure of the air alone must be

$$2.900 - 2.515 = 0.385 \text{ in. of mercury.}$$

As the air occupies the entire space of 598 cubic feet per minute and its pressure is only 0.385 in. of mercury and its temperature is 109 F, the volume of air handled and expressed at, say, 30 in. of mercury atmospheric pressure and 60 F would then be

$$598 \times \frac{0.385}{30} \times \frac{(60 + 460)}{(109 + 460)}$$

or

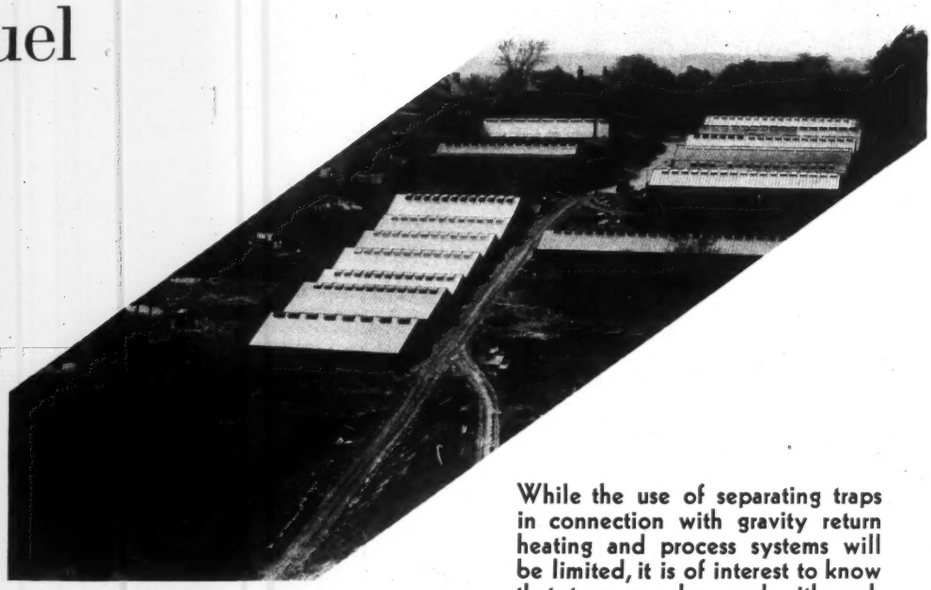
$$598 \times \frac{0.385}{30} \times \frac{520}{569} = 7.02 \text{ c. f. m.}$$

This represents the air leakage expressed at atmospheric conditions.

Volumetric tests of dry vacuum pumps handling mixtures of steam and air are not numerous and while actual figures are difficult to obtain, the indications are that the presence of the vapor reduces the apparent volumetric efficiency slightly. However, all such determinations by this method are intended to be used for estimates only—perhaps with a view to determining design factors for enlargements of plant or future installations. Figures obtained in this way will at least be of considerable help in preparing such estimates.

The figures herein have referred to reciprocating vacuum pumps. The volumetric line of the rotary-type pump has the same characteristics as that of the reciprocating machine.

Trapping Saves Fuel



A. W. Smith Company Greenhouse. The Boiler Plant Is in the Lower Left-Hand Corner. Large Fuel Savings Have Been Made by Proper Trapping

While the use of separating traps in connection with gravity return heating and process systems will be limited, it is of interest to know that traps can be used with such hook-ups. This article describes an installation where the use of traps has led to marked fuel economies, and shows possible variations of the arrangement which is used.

By T. H. Rea*

A CLOSED gravity return system as shown in Fig. 1 is a simple method for returning condensate to a boiler. The pressure drop between the boiler and the steam-using units *M, N* and *O* causes the steam pressure in the return line to be somewhat lower than in the boiler. As a result, check valve *C* in the return line remains closed until the static head of condensate in the vertical return line plus the return line steam pressure is slightly greater than the boiler pressure. Any additional condensate will increase the static head, overcoming the pipe friction, and condensate will enter the boiler.

The low cost and simplicity of the gravity return system may be offset by decreased efficiency of the heating and process units. Air must be eliminated from the coils or machines before steam can enter. When this is done by manual manipulation of valves, valuable steam is wasted. As soon as the units are blown down, air from other parts of the system or from the make-up water starts to accumulate and efficiency drops until the blow down process is repeated.

Due to the comparatively small difference in pressure between the steam supply and return lines, short circuiting frequently occurs when more than one machine or coil is connected with a gravity return system. If one machine uses considerably more steam than another, the pressure drop through the first machine will be greater. This will cause a back flow of steam from the return line to the machine. Although condensate can drain counter-current to the flow of steam, air will be pocketed and cannot escape until the pressure drop is decreased until it is approximately equal to the pressure drop through the other units. This falling off in efficiency frequently makes it necessary to increase the steam pressure in order to obtain proper temperatures. Extra fuel is required for the higher pressure and radiation losses from supply and return headers are increased, all of which costs money.

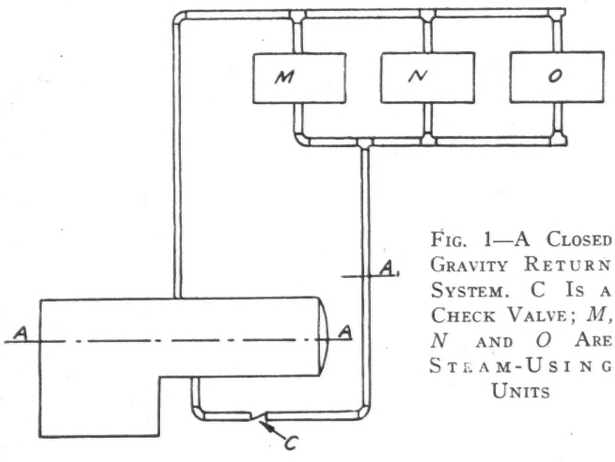


Fig. 1—A Closed Gravity Return System. C Is a Check Valve; *M, N* and *O* Are Steam-Using Units

*Armstrong Machine Works, Three Rivers, Mich.

Of course if a trap is installed on a cold machine or heating coil and the discharge is piped to an open tank or to the sewer, there will be sufficient pressure differential through the trap for satisfactory operation. Air as well as water will flow to the trap and the machine or coil should be able to operate at top-notch efficiency. Even though the heat units in the discharged condensate may be wasted, the increased production or better quality work from the trapped machine may be well worth the cost.

Separating Traps Used on Gravity Return Systems

It so happens that separating traps can be used on some gravity return systems without wasting the condensate and without using a pump or return trap. An installation of this kind is in the Pittsburgh greenhouse of the A. W. Smith Company. Due to the hilly terrain, most of the greenhouses in that section of the country are built on a side hill with the boiler room at the lower side of the range. A sketch of the heating layout is shown in Fig. 2.

Prior to the installation of separating traps, this greenhouse operated on a straight gravity return system. Since different kinds of flowers and plants are grown, it was necessary to maintain different temperatures for each house. Such a condition was ideal for short circuiting and a good many of the heating lines were filled with water and air. Considerable trouble was experienced with water hammer and fuel bills seemed unnecessarily high.

By installing a separating trap on each heating coil, live steam was prevented from reaching the return line. A maximum steam pressure of 10 lb. is carried on the boiler so a static pressure of slightly more than 10 lb. is the greatest required to force condensate back into the boiler. The elevation of the heating coil in house *M* is about 30 ft. above the water line in the boiler. Under these conditions, it is impossible for water to back up sufficiently far in the return line to interfere with the discharge from the lowest trap.

In order to prevent flash steam from building up back pressure in the return line and at the same time vent air from the system, a small valve was put at the high point on the return header; this valve is left partially open. As long as the valves in the separating traps remain tight, no live steam can escape from the return line. The radiation losses from the return line are more than sufficient to condense all the flash steam from the discharged condensate.

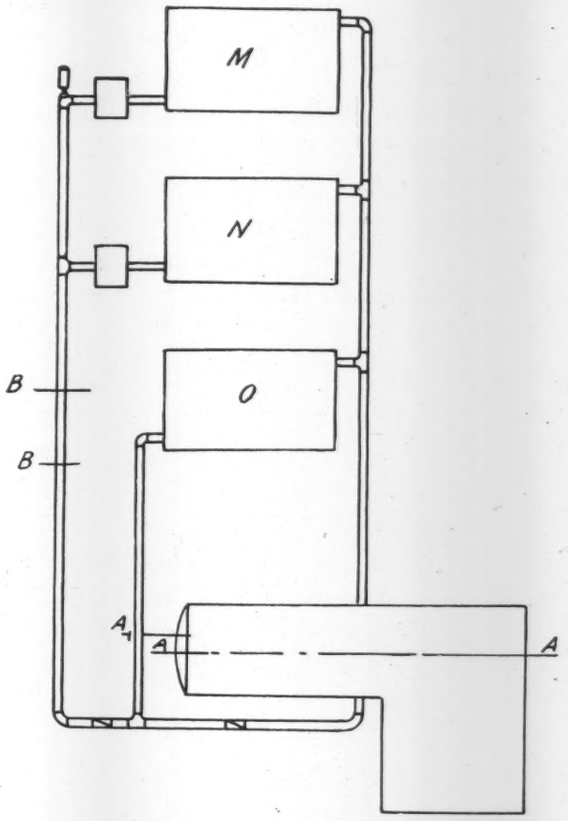


Fig. 3—Possible Hook-Up Where the Static Pressure in the Return Line Between the Water Level in the Boiler and the Lowest Coil Is Not Equal to the Boiler Pressure

Trapping Saves Fuel

With practically zero gage pressure in the return header, the pressure drop through the traps is enough to promote proper trap operation. The complete elimination of air from the coils makes it possible to obtain proper temperatures with lower steam pressures. Likewise, growing conditions are more easily controlled, which is of even greater importance than the decided decrease in fuel consumption. An unexpected advantage gained by using separating traps was a marked decrease in the amount of make-up water required. This was in part due to the elimination of steam loss through air vents on the heating coils.

On this job, the simplicity and low cost of a gravity

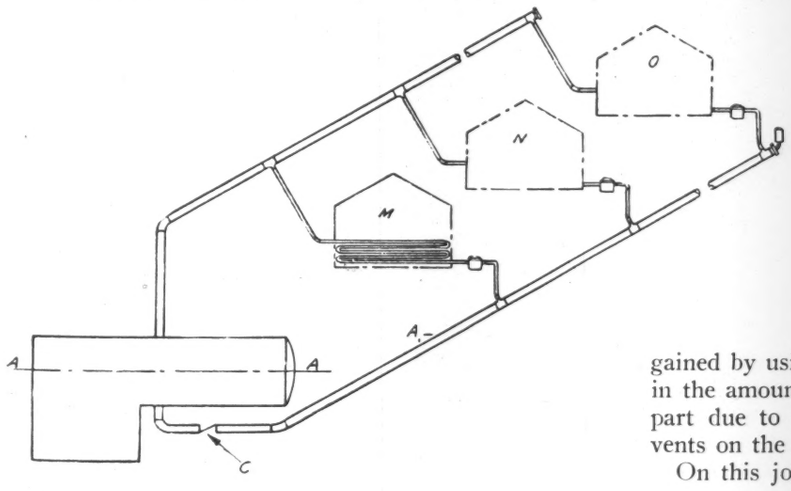


Fig. 2—The Heating Layout in the A. W. Smith Company Greenhouse, Which Is Located on a Side Hill

return system is coupled with the efficiency of proper trapping. From the standpoint of low cost, it is unfortunate that there are not more "side hill plants" that can put their boiler rooms sufficiently below the remainder of the plant to use a return system of the type described in this article. However, this method should have a fairly wide application for low pressure steam heating and process systems even when the difference in height between the water level in the boiler and the lowest coil is not very great.

Possible Arrangements

Fig. 3 shows a possible hook-up where the static pressure in the return line between the water level in the boiler and the lowest coil is not equal to the boiler pressure. The low coil or coils (as the case may be) can then be connected for straight gravity return while all coils or units above the high point for water in the return header could be trapped and the return header vented to the atmosphere. In this way, condensate and air can be drained positively from units *M* and *N*.

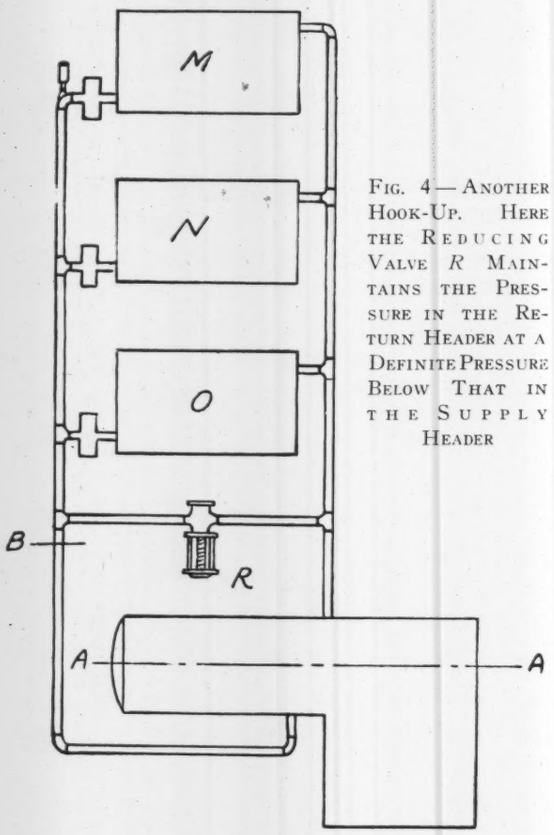


FIG. 4—ANOTHER HOOK-UP. HERE THE REDUCING VALVE *R* MAINTAINS THE PRESSURE IN THE RETURN HEADER AT A DEFINITE PRESSURE BELOW THAT IN THE SUPPLY HEADER

Another possible variation is shown in Fig. 4. Here the reducing valve *R* maintains the pressure in the return header at a definite pressure below that in the supply header. That is, if the pressure in the steam supply line were 30 lb., the reducing valve would keep the pressure in the return line at 25 lb. A 5-lb. change in the supply line pressure would cause a corresponding pressure change in the return line. Under these conditions, the water in the return line could not back up beyond point *B*, approximately 10 ft. higher than the water level in the boiler. With a positive pressure in the return header, it would be necessary to install a thermostatic air relief valve on the return line to prevent air from accumulating and building up the pressure in the return line beyond the 5-lb. differential.

While the use of separating traps in connection with gravity return systems will be limited, it is of more than passing interest to know that, contrary to the general belief, traps can be used in this manner.

Don't Under-rate Efficiency

Editor,
HEATING, PIPING AND AIR CONDITIONING.
Dear Sir:

PERHAPS no greater problem presents itself to the engineer than the question of when a machine has outlasted its period of usefulness. There are cases where the scrapping of plant is obviously necessary, even as there are cases where plant, though being obsolete, does not require renewal; but the majority lies between these two points—where a man with clear vision can ferret out the better choice between the two alternatives to a certain extent, yet in many cases cannot because of unknown factors beyond his control.

The obvious objective in any organization is to obtain the greatest return for a minimum outlay, yet how great is the number who can see this only in concrete terms of dollars and cents? Innumerable instances do arise where the expenditure of capital on equipment, over and above that which gives an actual return, secures additional returns even in greater ratio; but comparatively few engineers realize to what extent this avenue can be exploited.

I do not think that one can make any hard and fast ruling where the discarding of equipment is concerned, but I do believe that it is better to spend capital to ensure a peak in efficiency, rather than risk a loss in efficiency for the saving in expenditure that can be made. In my opinion, this general principle should be applied in any form of investment, and that in the long run it is the surest policy—provided the investment in the first place has had certain chances for success. To illustrate an example in the case of an apartment building; is the policy of cutting costs and risking efficiency wise—even if the net returns are increased? If, say, one of the heating boilers had been causing trouble and showed signs of wear, should one risk patching it for another year's working? I would consider the life of that boiler, and if it were anywhere near the average usually obtained, it would be scrapped without hesitation. It would be looking for trouble to expect to get more, even with numerous examples of boilers having lived long past their allotted span. Interrupted or inefficient service to my mind is usually inexcusable. Taking another form of plant such as a factory, I would consider the same general principle in this way. Conditions may be good over a long period where the demand remains steady—in which case men are often tempted to forget efficiency when returns are good, but the day can always come when the true test in the falling off of demand induces competition, when prices are keen. "The wise general," said Napoleon, "never under-rates his opponent's strength." I would add an analogy that "the wise engineer should never under-rate Efficiency."—*Cyril L. Tilburn,* Sassoon House, Shanghai, China.

A Review of Dust

The object of this article is to survey the great kingdom of dust, first in a general way and then more specifically with reference to dusts encountered by the heating, piping, and air conditioning engineer. Its purpose is to explain why the engineer should know about dust, what he should know, where dust is located, the character of dust, how it is measured or counted, and how the engineer is concerned about dust in its various forms. Future articles will present detailed information.

By L. B. Spafford

DUST is one of the most intimate associates of our lives. The menace of dust was recognized in the writings of the ancients; three hundred years before Christ, Hippocrates complained about dulling his knives when performing autopsies on stone workers due to the amount of dust in the lung tissue. Agricola called particular attention to the fact that the women in the Carpathian mountains outlived their husbands due to the high death rate of the miners, women often being married as many as six or seven times. The elder Pliny devised a mask to protect workers in certain industries. Dust is the material from which a large part of our manufactured products are made. Dust is held responsible for a disease that is more deadly than the greatest war known to history.

Why the Engineer Should Know About Dust

Dust is of concern to the heating, piping, and air conditioning engineer, for it is removed, conveyed, conserved, or separated for physiological or manufacturing reasons by means of air which is propelled by fans and blowers, driven by motors and engines through hoods, ducts, and pipes, filters and washers which are common tools of the engineer in the heating, piping, and air conditioning fields.

The engineer should know about dust because it has a profound effect upon human beings in health and in sickness, in surgical operations, in affording protection from ultra-violet rays of the sun, in commerce and manufacturing, in its effect upon manufactured products, in agriculture in conveying seeds and spores to barren spaces, in fertilizing the ground, in the simoon or dust storm, in making light visible, in affecting weather, in making beautiful sunsets, in producing fogs, in its decomposition on hot surfaces, in its de-ionizing effect, and for many other reasons.

The engineer should know about dust because many manufactured products are made of dust particles. He should know about dust because he has it in his power to make dusts useful, to remove those that are harmful, to handle dust economically, to separate dust particles as to size, to handle dusts so as to reduce manufacturing costs, increase production, provide better working conditions, recover valuable materials, safeguard against explosions, lessen fire hazards, remove waste and refuse, protect machinery from dust, convey materials economically, and remove fumes, gases and obnoxious odors.

What the Engineer Should Know About Dust

In the engineering of occupied buildings and in manufacturing, the engineer should know many things about dusts; their compositions, weights, sizes, shapes, physiological effects, their crystalline structures, whether the particles are granular, globular or flocculent in form, the dust concentration as affected by the gas velocity, the rate at which dust settles, the volume of gas that carries dust, the temperature of the carrying gases, the humidity, the gaseous envelope that surrounds certain dust particles, and how readily they adhere to solid surfaces. It is important that the engineer know of methods available for collecting and conveying the dust-carrying gas, for sep-

arating the dust from the gas, and for the handling or use of dust so separated. He should know how to select, design, operate and maintain these various pneumatic systems so that they will function properly.

What Is Dust?

One dictionary states that dust consists of earthy or rocky material so fine that it can be borne away by the wind. Webster's informs us that dust consists of fine particles of earth or other materials so comminuted that they may be raised or wafted from the earth.

But there are many kinds of dust which do not conform to these specifications. Dust is not determined by buoyancy in ordinary air as the dictionaries would lead us to believe, but rather by the detachment from or lack of cohesion with associated particles and by such shapes, sizes, and weights as will permit them to be borne by natural air currents or blasts of air produced by mechanical means.

Where Is Dust?

Dust is everywhere, indoors and out-of-doors, in the city and in the country, on the mountains and on the plains, in polar regions and in the tropics, at the bottom of the ocean and even in the infinite spaces beyond the earth. Strictly speaking, there is no such thing as fresh (pure) air except that which is produced by mechanical means, and that is only relatively pure. The "fresh" air that comes in at a city window from time to time may be comparable to the "fresh," sparkling, spring that flows through a dumping ground.

If one should insist on surveying the whole kingdom of dust, he would have to start with particles smaller than can be seen through any microscope,—particles of dust inconceivably small worn from a coin in the pocket until the coin is smooth, for instance, or the particles which may be given off for many years by a grain of musk without perceptibly lessening its weight, for it is the impact of such ultramicroscopic dust particles on olfactory nerves which is responsible for our sensation of smell.

We have no means of learning how many of these exceedingly small dust particles there are in a cubic foot of air for they can not be detected by any instrument devised by man. We may learn some day that these infinitely small and undetectable particles of dust in air are really the "unknown stranger" in air which many physiologists, physicians, and engineers charge with the responsibility for a rise of respiratory diseases as soon as colder weather causes human beings to keep themselves inside of dusty buildings. Dust counters do not start to function until dust particles are at least 1/100,000 of an inch or ¼ micron in size. Particles one micron in diameter usually are readily detectable. If a cubic inch of coal were broken into smaller cubes, ½ micron square, that one cubic inch of coal would produce 1,250,000,000,000 particles of dust.

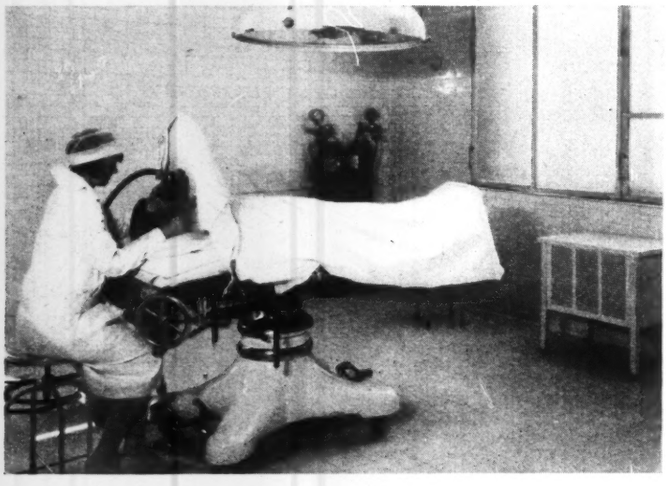
The Kinds of Dust with Which the Engineer Is Concerned

But the dusts with which heating, piping and air conditioning engineers are most concerned form but a small part of this great kingdom of dust. They are the dusts that are detectable by man. They are those which we think may have some readily observable effect upon health and comfort, which will help or hinder in manufacturing, which must be conserved because of their value, which must be removed because they are obnoxious, which must be separated as to their size in order to improve quality or maintain the standard of a product, or which must be conveyed.

The largest particle of dust in this field of what we might term physiological and obnoxious dusts probably would be 72 times the diameter of the smallest particle. They might range from 1/25,000 of an inch to 72/25,000 of an inch with average dust particles 3/25,000 to 4/25,000 of an inch in size. Dust particles, however, are usually measured in microns. A micron is 1/1,000 of a millimeter or approximately 1/25,000 of an inch.

Objectionable Dusts

In that province of the kingdom of dust known as obnoxious, objectionable, or unhealthful dust there may be grains of pollen, bits of vegetable fiber, hairs,



Filtered Air Is Supplied to This Operating Room

carbon, particles of earth, ashes, particles of horseshoes and harnesses as well as automobile tires worn off on the cobble stones and pavements, threads of cotton, coal dust, shreds of wool or silk, animal excreta, smoke, water-dust, clay, stone, and materials of infinite variety and often riding on these particles are bacteria, fungi, and other living germs some harmful or beneficial or indispensable to human beings and animals; some harmless, some harmful, some beneficial or even necessary in manufacturing.

One classification of dust might be as follows: Dusts from outdoors which are removed from buildings because of their effect upon human beings or animals (these have been mentioned heretofore); and manufacturing dusts; these may be marble dust, granite dust, mercury dust, fumes from smelting ores, wood dust from sanding machines and other wood operations, silica dust, zinc oxide, fumes from plating baths, dusts from brass foundries, magnesium oxide, aluminum, chromium, iron and other oxides, particles and pigments and oils from spray painting, metals from grinding or buffing machines, fly ash, calcium dust in a Portland cement plant, silica dust in granite-cutting plants, carbon dust in coal mines, vegetable dust in a cotton manufacturing plant, and metal and other dusts in a silver polishing plant.

Useful Dusts

Some of the dusts that are the foundation of manufactured products or are usable in dust form are: Portland cement, pyroxlyn powder, bakelite powder, rubber, silica, charcoal, granulated slate, powdered coal, bone ash, soap powder, insecticides, volcanic ash, grains such as wheat, oats, rye, barley, and corn, synthetic coffee, talcum powder, toilet powder, gypsum, paints, dyes, kitchen cleanser, tungsten, litharge, wood flour, chicken feed, powdered milk and so forth.

Size and Characteristics of Air-Borne Impurities

By W. G. Frank*

The widespread interest in air conditioning is based on the recognition and general acceptance of the fact that, for maximum comfort, the air must be properly cleaned and correctly distributed, as well as heated or cooled and humidified or dehumidified. All these factors or processes, their correct relationship, and the reasons for their importance in air conditioning are the subjects for much discussion and study. Comparatively little thought, however, is given to the basic reasons for cleaning air and to the intricate problems that are involved. Everyone is, of course, familiar with, and often annoyed by, the dust and dirt in the atmosphere, but specific data on the nature and behavior of air-borne impurities are not readily available, as they are scattered through the technical and scientific literature of many countries. **The air conditioning engineer who is frequently faced with the need of discussing and vizualizing dust conditions will, therefore, find this review of value.**

IT IS common knowledge that the atmosphere contains numerous fine particles from many sources which float about freely, and settle when and where they please. But little do we realize that we are surrounded by a veritable ocean of particulate matter ranging in size from the all too noticeable dirt and grit to particles of unbelievable smallness. Their number is as large as their size is minute. Most of them are visible only under the microscope, or the ultramicroscope, and many reach beyond into invisibility.

The study of dust is an interesting one. The field is so vast and touches so many phases of science, that many volumes would be required to cover it completely. Therefore, this discussion will deal briefly with only a few of the most important characteristics of suspensions or disperse systems, sometimes called aerosols or dispersoids. The main purpose is to present a convenient classification of the sizes of dust particles together with other related data of interest.

Origin of Dust Suspensions

Particles of very small dimensions have a relatively large surface area as compared to their weight. Thus, when suspended in a gaseous medium, they are freely carried by virtue of the viscosity of the gas. The system thus formed is not in equilibrium and is subject to constant changes. The degree of instability depends upon the size and properties of the particles, the medium in which they are suspended and the effect of outside forces such as gravitation, centrifugal force, convection currents or magnetic force.

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Suspension of particles in gases may be originated by:

1—Mechanical disintegration of solids.
Cutting, crushing (street dust), grinding, milling (flour, cement, etc.)
2—Combustion and explosion.
Reduction of solid and liquid fuels to ashes, fumes and smoke (volcanic dusts).
3—Mechanical dispersion of liquids.
Spray evaporation (powdered milk).
4—Condensation of gases and vapors.
Formation of rain from water vapor, condensation of metallic vapors.
5—Chemical reaction.
Production of dense white smoke by mixing silicon chloride and ammonia (smoke screens in warfare).

Most of the impurities found in the atmosphere are the result of processes *1, 2* and *4*. In addition there are numerous particles of animal and vegetable origin, such as bacteria, pollen, mold spores, lint, etc.

These impurities may be of a solid or liquid nature. If they are liquid—such as rain drops—they may take a spherical form. Solid particles, however, are not usually round. Their form depends upon the structure of the material, which may be crystalline or amorphous, and upon the process by which they were generated. Usually dust particles are geometric reproductions of the original substance. Thus, microscopic wood dust is fibrous, having the appearance of splinters; minute silica particles show the same irregular form as large stones.

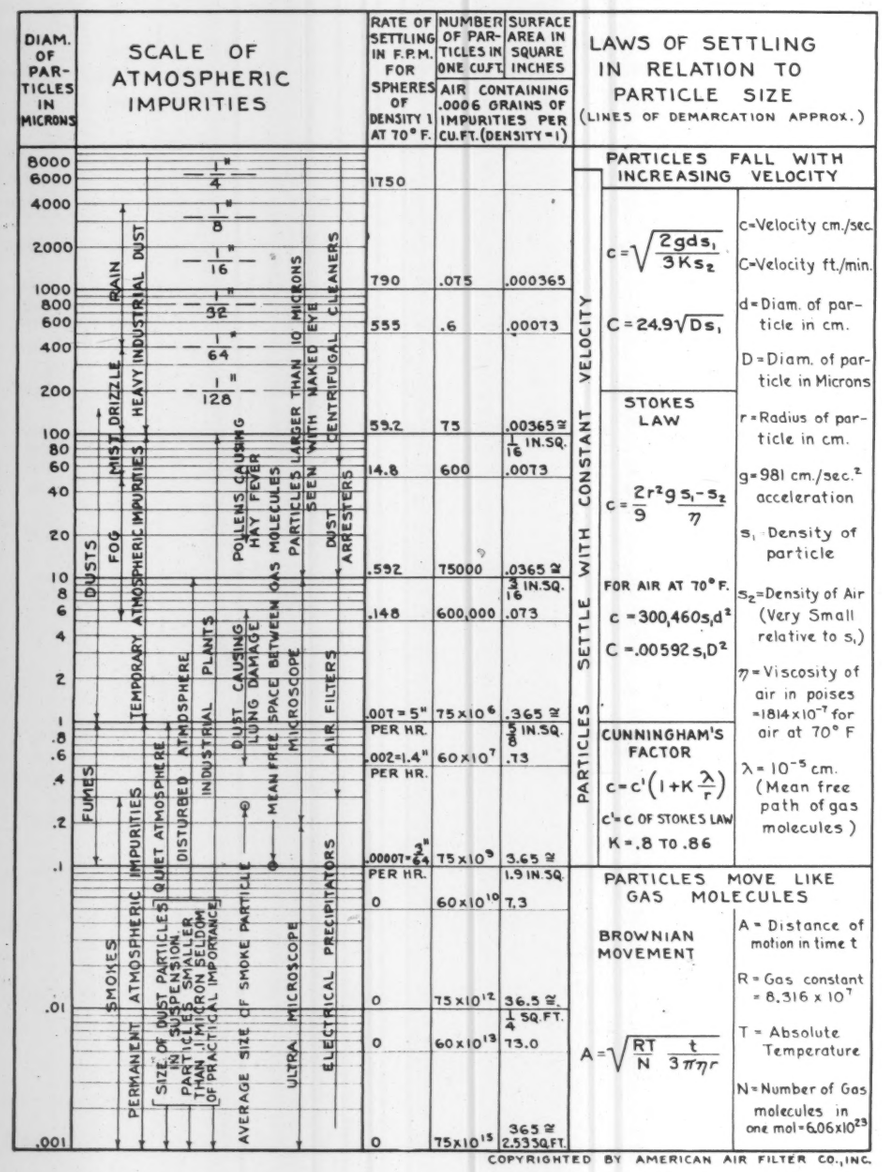
Measuring Dust Particles

Arithmetical Mean: To indicate the size of such particles, their diameter is usually given. Only perfectly round particles have a true diameter; for those of irregular form the mean of length and width, or length, width and height, takes it place. The diameter of such dust particles may then be added and the average diameter can be found as the arithmetical mean. Particles which are considerably larger in one direction, for instance fibers and bar and rod shapes, require correction factors when the laws of settling are applied.

Surface Mean Diameter: When working with dust in the form of a fine powder, it is often convenient to figure with the *surface mean diameter* of the powder. This can be easily calculated after the arithmetical mean diameters of the particles have been determined. The value obtained represents the diameter of uniform spherical particles which, for the same weight of dust sample, have very closely the same surface area as the original particles. This surface mean diameter (d_s), is obtained from the equation

$$d_s = \frac{\Sigma d^3}{\Sigma d^2}.$$

The surface mean diameter is always larger than the arithmetical mean and is a measure of the specific surface of dust. The latter in turn determines the ability



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SIZE AND CHARACTERISTICS OF AIR-BORNE SOLIDS

Compiled by W. G. Frank

of the dust to remain suspended, and has an important bearing on its chemical activity.

Volume Mean Diameter: In conformance with the foregoing, the *volume mean diameter* of dust particles may be established, by using the equation $d_v = \frac{\Sigma d^4}{\Sigma d^3}$. The volume mean diameter indicates spherical particles having a volume equal to that of the original dust particles, for the same weight of dust sample.

Dust, Fumes, Smoke: When speaking of atmospheric pollution, it is customary to classify the impurities in three broad groups; there are no exact boundaries and the transition from one group to the next is gradual. The distinction in classes, while not scientific, is convenient because it is quite descriptive of the origin of the particles and their manner of moving. Dusts settle without appreciable agglomeration; fumes tend to aggregate; smokes, to diffuse.

A Useful Dust Chart

The chart published herewith shows a scale of these classes of atmospheric impurities, and locates some of the dusts of everyday importance within this scale. The particle diameters are expressed in microns and arranged on a logarithmic scale. One micron is equal to 0.001 millimeter, or approximately 0.00004 in. It is not surprising that for atmospheric impurities such a minute unit of measurement is necessary. A micron is really comprehensible only to the eye of the microscope. Even the human hair, proverbially the last word in smallness, has a diameter of 60 to 150 microns, or on the average about 100 microns.

As a further comparison the size of openings in screens, as used for dust analysis, is given in microns:

140 mesh	104 microns
200 mesh	74 microns
300 mesh	47 microns
325 mesh	44 microns

The chart also shows the rate of settling for particles in still air. It is assumed that the particles are spherical and of unit density and thus may represent water drops. The next two columns give the number and the surface area of particles in one cubic foot of air for a certain dust concentration. It is of considerable interest to note how rapidly, for a fixed concentration, the number of particles increases as their size decreases. The dust concentration in the chart is 0.0006 grains per cubic foot which is an average of a number of dust-weight determinations in the heart of several large cities. Atmospheric dust has a density between 1 and 2 and is irregular in form. To simplify matters we will assume that the particles are of uniform spherical shape and have unit density.

A cubic foot of this city air would then contain 75 particles of 100-micron, or 75,000 of 10-micron, or 75,000,000 of 1-micron diameter, to make up the weight of 0.0006 grains. Many observations have shown that the median size of the dust particles normally present in the atmosphere is below 10 microns. Their number in a single cubic foot of city air must, therefore, exceed 100,000 and may be many times greater.

This is confirmed by dust counts made with instruments designed especially for this purpose. The number of particles counted in one cubic foot of air varies

with the dustiness of the air, but the dust count is basically dependent on the manner in which the dust counter collects the particles on glass slides and upon the degree of magnification. Dust determinations made with the Hill dust counter in a number of large cities have yielded an average of 9,700 particles per cubic foot; an average of 500,000 particles per cubic foot was counted during an investigation of the air in New York by means of the Owens jet apparatus; with the Aitken counter many million particles per cubic foot may be found in ordinary air. (Aitken found 3,736,000 particles per cubic inch in Glasgow.)

Laws for Settling of Dust Particles

The chart is concluded with the laws which govern the movement of particles and the range of dust sizes in which the laws are effective. The standard law of fall is not adequate for particles less than about 7 millimeters in diameter (¼ in.) and is therefore of no practical importance as far as atmospheric impurities are concerned. Below this particle size the influence of the viscous friction of the air becomes increasingly important and tends to lower the rate of fall. Comparatively little is known concerning the exact rate of fall for particles from 7 millimeters (7,000 microns) down to 200 to 100 microns, where the law of Stokes comes into effect. Research in this range of particle sizes is under way, and the formula by Prockat, given in this chart, was developed from experiments in 1929. Particles of this range of sizes, with the exception of rain drops, are practically never found in the atmosphere. Even if they are temporarily carried in the air by particularly high winds, they settle so rapidly that they do not present a problem.

The great majority of impurities that are technically important range in size from 200 microns down to approximately 0.1 micron. About 30 years ago, G. G. Stokes developed the law which states that particles of such size will fall at a uniform velocity, the resistance of the air balancing the downward pull by gravitation. This constant velocity will be directly proportional to the square of the diameter of the particles. For different substances, the heavier particles will fall faster, or if at equal velocities, will be smaller than lighter particles.

The Stokes law holds exactly only for perfectly round particles, but numerous experiments have demonstrated that particles which do not differ greatly from the spherical form, such as fly ash, silica or carbon, follow the law very closely. For dusts which are shaped like discs, rods, etc., a correction factor must be applied. These shapes will fall in the position in which they encounter the maximum air resistance.

It is evident that the upper limit of particle size for which the Stokes law is strictly applicable depends on the density of both the falling body as well as the carrier medium. The upper limit for water droplets has been established as 200-micron diameter, while for fly ash it is reached at about 100 microns.

In this class of dust must be counted air-borne pollens which are thrown off by grasses, weeds, trees and

flowers, during certain seasons. They are the principal cause of certain allergic diseases, notably hay-fever, and bring untold suffering to many people who are susceptible to this type of protein irritation. It is estimated that more than three million people in the United States are subject to this seasonal affliction. Among the most frequent causes for hay-fever are these pollens:

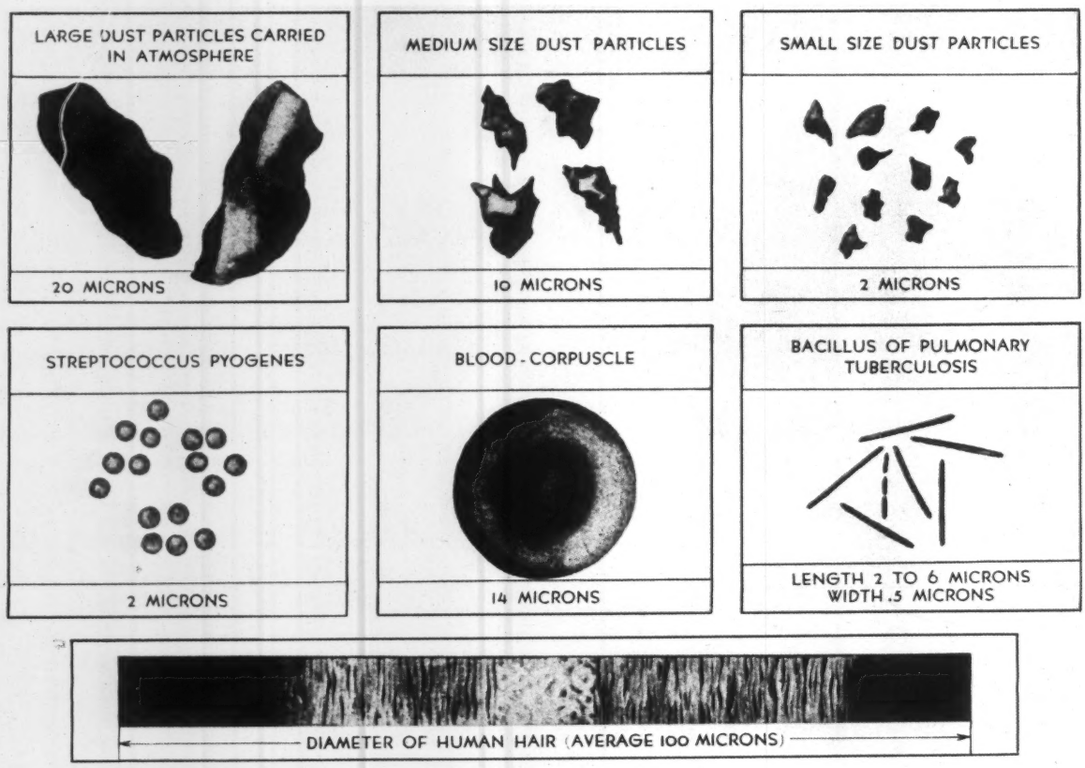
Ragweed	18 to 24 microns diam.
Grasses, such as Timothy grass, Johnson grass	40 to 50 microns diam.
Tree pollens, such as maple and oak	36 to 52 microns diam.

Fumes

The boundary line between dusts and fumes (clouds) is not very definite, as between 10 and 1 micron we find fumes as well as particles that are ordinarily called dust, such as starch, silica, carbon, etc. Of these dusts the sizes from 6-micron down are of special hygienic significance, as in breathing they reach the depths of the lung. If the concentration of the dust is sufficiently high, between 40 and 60 per cent of the particles from 6 to 0.5 microns are retained in the respiratory tract and the depths of the lungs. Such large amounts of dusts, if inhaled consistently over a long period of time, will cause lung damage. The seriousness of the disease depends on the chemical composition and the size, shape, and nature of the dust. Workers in many trades are exposed to this health hazard, among them brass-workers, grinders, painters, stone-cutters, spinners, furriers and hatters. Absolute protection against these dusts is often difficult to provide, but industrial management must strive to improve working conditions in these trades to the point where the dust hazard becomes negligible.

Fumes, especially those resulting from chemical processes such as ammonia chloride, sulphuric acid and zinc oxide are very unstable. The individual particles grow by aggregation to the order of 1 to 10 microns, and consequently settle out steadily in still air. The rate of fall, as shown on the chart, is so slow that their settling may be retarded or even prevented by convection currents.

If the particle has a diameter of 1 micron or less, it gradually comes under the influence of the movement of the gas molecules, resulting in what is known as the *Brownian movement*. The mean free path of the molecule is about 0.1 micron, and particles approaching this dimension will find less resistance in their motion through the air than indicated by Stokes law. They will settle faster and the true velocity is obtained by using Cunningham's correction factor. Such small particles, however, will practically never settle in the ordinary atmosphere where convection currents are constantly at play.



Comparison of Dust Particles with Various Small Organisms

The particle diameter of 1 micron may well be called a critical size for settling in the atmosphere. Particles above 1 micron will eventually settle out and have therefore been called *temporary impurities*. The finer particles, below 1 micron, will remain in suspension as *permanent impurities*. Due to gravitation, they tend gradually to seek lower levels, but will settle only if they aggregate to sizes larger than 1 micron.

Smoke

Below 0.27 to 0.1 micron is the region of smokes. The particles are so small that the impacts from the ceaselessly-moving molecules drive them to and fro in a continuous Brownian movement. They may travel in

any direction, and the effect will be that they gradually diffuse. The amplitude of their movement in a given time may be determined, but the figure has little practical value. Here the limit of the microscope is reached as the particles are small compared with the wave-length of light. They must be viewed through the ultramicroscope which renders them visible down to a size of 0.001 micron, although their shape cannot be seen. Larger particles simply appear brighter than smaller ones. These are the particles which serve as nuclei for the condensation of water vapors.

Explosive Dust Suspensions

The fineness of the dust or degree of dispersion is of considerable importance in considering explosion hazards. The finer the particles of a dust cloud are, the larger is the area of the dust particles and the more readily a chemical reaction with the oxygen of the air takes place. Thus inflammable dusts in suspension may give rise to terrific explosions and this industrial hazard has received special attention and systematic study in recent years.

How greatly the surface area of a given weight of dust increases with decreasing size of the particles, is shown in the last column of the chart. The dust concentration of 0.0006 grains per cubic foot is by far too low to form an explosive mixture, even if the nature of the dust were such that it would be possible to ignite it. As a relative comparison, however, the increase in surface area with diminishing particle size is surprising even at such a low concentration. In addition to the size of the dust, the shape and surface condition of the particles affect their inflammability. The amount of ash and volatile matter, as well as moisture content and chemical composition, are also of importance.

The most frequent dust explosions occur in coal mines where fine coal dust is often mixed with inflammable gases. But there are other dusts that are to a greater or lesser degree explosive, and in fact all carbonaceous materials, and other substances that will burn, must be included among the dangerous dusts. To this classification belong sugar, starch, cocoa, rice, cork, wood flour, malt, grain, wheat flour, tea, paper, spice and many other common industrial dusts.

It was originally supposed that dust clouds explode only when combustible gases are present and that they propagate the explosion through the volatile matter resulting from the burning dust particles. It is known now that at the proper dust concentration a spontaneous reaction takes place between the finely-divided dust particles and the oxygen molecules of the air. This occurs so rapidly that the pressure therefrom is fully capable of igniting the remaining unburned dust particles.

Fighting Atmospheric Pollution

The properties of dust suspensions and the size, nature and movements of the small particles which make up suspensions in the air, represent a vast field for analytical and experimental reseacrh. Science knows a great deal about this field, but there are questions which have so far been answered only empirically. For instance the ability of dust particles to adhere to surfaces of cloth, liquid-coated baffles and other media is governed by laws which are extremely difficult to formulate. These phenomena are of great importance in the design of air cleaning equipment, but practical experience has so far remained the only guide in applying them to the problem of dust elimination.

Thus we are brought to the question of what may be done to protect ourselves against the avalanche of dust, fumes and smoke. We may never subdue and it is perhaps futile to concern ourselves about the host of ultramicroscopic impurities, but we must not cease our efforts against the essential portion of air pollution that is revealed in the microscope, that shuts off a goodly portion of sunlight and affects our health, damages our machines and contaminates our materials in process.

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Industrial Heat and Power

MR. BARRY'S article[1] on high pressure steam in the industrial plant published in the July issue is an excellent exposition of what can be accomplished in industrial power. The author, of course, is dealing with the question from the power end, but high pressure steam also has extensive possibilities in direct process use.

From the power generator's point of view it is desirable to have the back pressure as low as possible. From the exhaust steam consumer's point of view the reverse is frequently the case.

In multiple effect evaporation the steam consumption may in some cases be cut in half by raising the first effect pressure from 5 to 10 lb. gage to 100 or 150 lb. gage. With other apparatus, such as heaters and dryers, the capacity may be enormously increased by using higher pressure.

The supply of power and heat in the industrial plant should be regarded as a single problem and the intent should be to make the product at the lowest possible cost, rather than to develop power in great quantities and at low cost. In the majority of industrial plants revision of the process apparatus will save more fuel at less capital cost than revision of the electric power generating apparatus. A boiler plant would probably have to be altered to accomplish the best results in either the power generating or heat consuming fields.

This additional thought is in no way a criticism of Mr. Barry's paper, since he assumes that before developing a high pressure boiler plant the consumption of steam has been made as low as possible in the process end. This is an assumption, however, which would not apply in many cases.—*Wm. F. Ryan,* Stone & Webster Engineering Corporation, Boston, Mass.

[1] Page 547, Heating, Piping and Air Conditioning, July, 1931.

THE NEW ST. LOUIS GLOBE-DEMOCRAT BUILDING

Newspaper Plant Air Conditioned

for Efficient Production

Information Furnished by

Joseph A. Osborn*

WITH plans adapted from the experiences of some of the most successful newspapers in the nation, and with many innovations especially applied to its own problems, the St. Louis Globe-Democrat completed in November a new building which has revolutionized newspaper production for that publication. The plant occupies almost an entire city block.

Radiation

The major portion of the building is heated by means of direct radiation. Due to the fact, however, that it was necessary to humidify as well as heat the main press room, this portion of the building is served by an indirect system.

Steam for the building is purchased from the local power company, which maintains a line pressure between 150 and 175 lb. The heating service enters the building in the sub-basement. Reducing valves are provided in order that the required heating pressure can be obtained.

The direct radiation throughout the building is of the legless type, suspended on hangers.

Temperature control is accomplished by automatic thermostatic valves, installed on alternate radiators throughout the building. A typical installation is shown in Fig. 2, which indicates clearly the installation of one hand valve and one thermostatic valve.

Use of Condensate for Hot Water Supply Conserves Heat

The hot water requirements for the building are furnished from two large hot water generators, located in the basement and shown in Fig. 1. One of these generators is sufficient to supply the hot water requirements, but two were installed in order to insure continuity of operation. The hot water generators are served from a line carrying approximately 40-lb. pressure. This steam line is provided with a diaphragm valve operated from a thermostat in the hot water tanks, for maintaining a predetermined and constant temperature of 160 F.

In view of the fact that the building is served from the power company and ordinarily the condensate from the heating system is discharged into the sewer, there has been provided a condensate pump, shown in Fig. 1. The function of this pump is to introduce the condensate from the heating system into the hot water supply of the building.[1]

In this way a large amount of water and heat are saved and it is anticipated in cold weather, when the condensate returns in large quantities, that there will be no necessity of using live steam for heating the domestic water.

Press Room Is Humidified

Static electricity is a problem in unconditioned press rooms. When a press must be re-threaded because of a break in the paper due to static, time is consumed. And time is one of the most valuable factors in newspaper production.

Therefore, there has been installed in the sub-basement an air washer having a capacity of 80,000 c.f.m. The heating coils of the air washer are automatically controlled from a thermostat located in the press room to maintain an even temperature.

The air washer is provided with two sets of sprays, one controlled by a manual control valve and the other controlled by an air-operated diaphragm valve which is controlled by a humidostat in the press room. A constant temperature of 70 F and a relative humidity of 58 per cent is maintained in the press room. With evaporative cooling, a differential temperature of 10 F is expected in summer. Fig. 3 shows the continuous oil-cleaned air filter, the air washer, the fan and the circulating pump, before the complete housing was installed.

It is the intention to recirculate approximately 65 per cent of the air and admit 35 per cent fresh air. All regulating dampers are controlled by thermostats in order that they will function automatically.

Due to the large quantity of air handled, the circulating ducts are very large. Fig. 4 indicates a typical installation of one of the large recirculating ducts from the press room. It will be noted that it was necessary to construct these ducts shallow and wide in order to allow sufficient head room for storage purposes.

Individual Exhaust Systems

In addition to the ventilation of the press room, there has been provided a ventilating system for the stereotype department, the dark rooms and the etching room. In the stereotype department suitable ducts and vent openings were installed, leading to an exhaust fan in the pent house, which is operated by remote control from the stereotype room. Its function is to remove all gases and fumes incident to the operation of the stereotype equipment.

Due to the presence of acid fumes in the etching department, the fan and all ducts were constructed of 18 per cent chromium—8 per cent nickel alloy. The discharge from the fan passes through a 12-in. vitrified clay duct, extending from the etching room to the roof. A typical installation of exhaust ducts and registers in the stereotype department is shown in Fig. 5.

*Consulting engineer, St. Louis, Mo.

[1] For a detailed description of a similar system see the article "Condensate from Heating System Used as Hot Water Supply for Building," by R. M. McQuitty, which was published in the September, 1930, HEATING, PIPING AND AIR CONDITIONING, page 762.

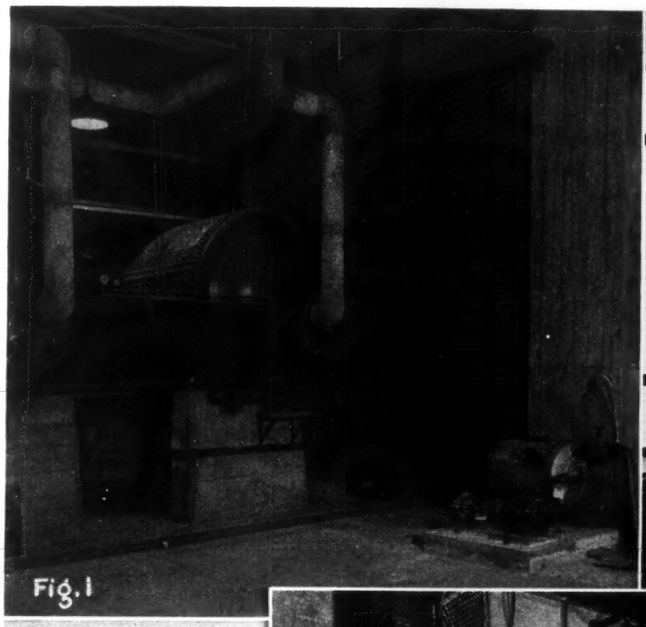


Fig. 1

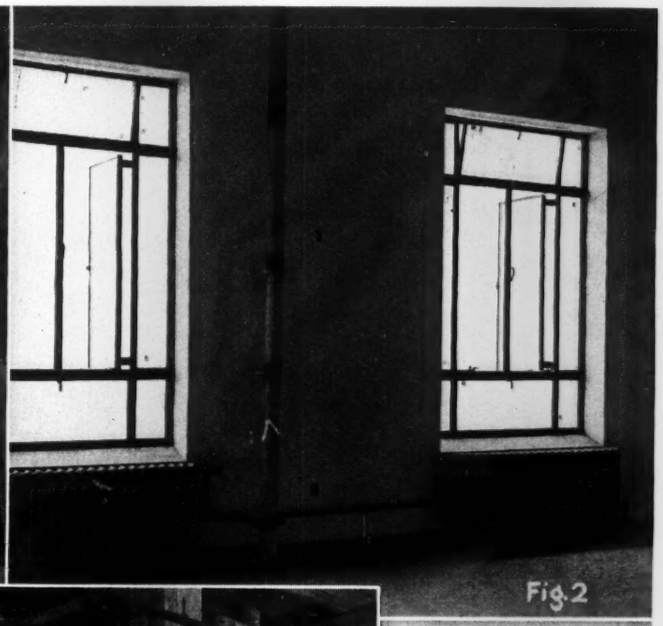


Fig. 2

Fig. 3

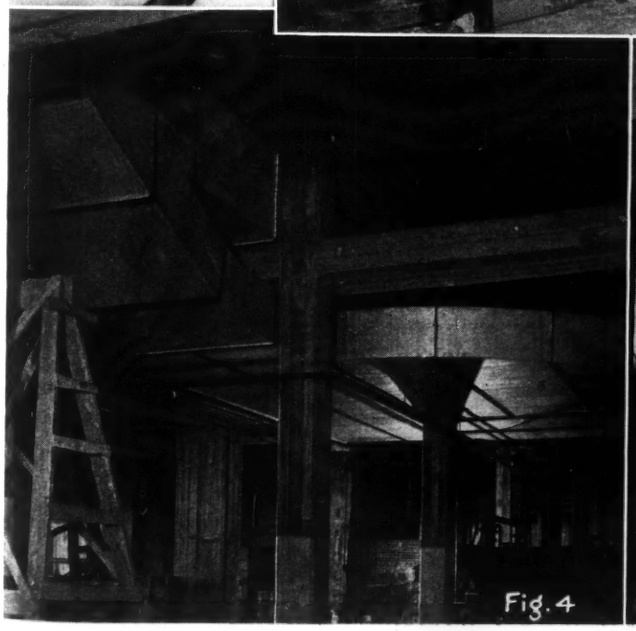


Fig. 4

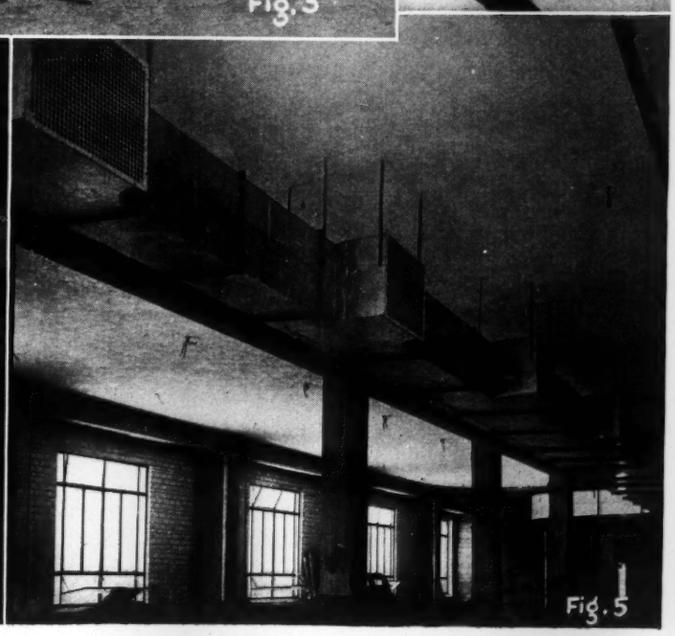


Fig. 5

Pre-Cooling Fruits and Vegetables with Circulating Air

By Clarence E. Baker*

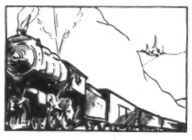
Reducing the temperature of perishable fruits and vegetables is one of the greatest problems in the economical handling and shipping of such produce. The value of pre-cooling and the reasons for doing so are discussed in this paper and a description of the methods used is given. Proper pre-cooling depends to a large extent on the correct circulation of air.

ONE of the greatest problems involved in the economical handling and shipping of quickly perishable fruits and vegetables such as cherries, plums, peaches, tomatoes and other kinds of produce is that of quickly reducing the temperature of the produce to a point where rapid deterioration will not take place. The tender fruits are subject to various rots and molds that grow very rapidly at temperatures above 50 F. Frequently cars of produce that have been shipped for long distances, even when properly iced, reach their destination in a worthless state as a result of lack of adequate cooling to retard the growth of fungi.

The Value of Pre-cooling

The checking of the ripening process, also, must be considered. Without the destructive action of rots and molds, a few days upon the road at high temperatures may cause excessive ripening of the produce to the extent that its quality is greatly reduced, even though it may be free from decay. Temperature is exceedingly important as is shown by the work of Gore (*1*)[1], who found that with forty different kinds of fruits the respiration increased an average of 2.376 times for each 10-C rise in temperature. With some varieties of peaches, strawberries and early apples the rate of increased respiration was in excess of the above figure.

It is exceedingly difficult, even with properly iced refrigerator cars, quickly to reduce the temperature of the load to a safe level. Ramsay (*2*) studied the temperature conditions of a car of apples in transit between Wenatchee, Wash., and Chicago, Ill. Under these conditions it required 114 hours and four re-icings to reduce the average temperature of the car from 62 F to 45 F and four additional days were required to reduce the temperature to 40 F.



A car of strawberries shipped from southwestern Missouri to St. Paul, Minn., was observed by Ridley (*3*), who found a difference of 10 F between the fruit in the top tiers of the car and that of the bottom containers.

Studies by Lloyd and Newell (*4*) indicate that it is difficult to secure satisfactory reductions in temperature during short periods of transit when warm fruit is loaded into iced cars even when salt is used upon the ice to secure lower temperature conditions within the car. They especially emphasize the importance of floor racks to increase the efficiency of air circulation and the loading of packages in such a way that air movement is not obstructed.

Heat Generated by Fruits

These statements indicate the value of pre-cooling of perishable fruit before loading or immediately after loading. The sooner the temperature of the fruit is reduced the greater will be the effect upon its ultimate "keeping quality." Fruit after harvest actually increases in temperature through self-heating. This condition was investigated by Overholser, Winkler and Jacob (*5*) who found that fruit exposed to the sun even while still attached to the tree was from 7 to 10 F warmer than the air temperature. Griffiths and Auberry (*6*) computed the amount of heat generated by various fruits and report that sound apples at 68 F generate 0.012 calories per kilogram per second. This equals approximately 3,740 Btu per ton of apples in 24 hours. More tender

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[1] See bibliography.

CALIFORNIA ORANGES BEING TAKEN TO A PACKING HOUSE TO BE MADE READY FOR SHIPPING

fruits generate greater amounts of heat, peaches developing nearly twice as much and raspberries about five times as much heat as apples. Gore (*1*) found peaches to increase 1.26 F in 10 hours when stored at 50 F and 6.57 F when held at 86 F.

These figures emphasize the importance of rapid cooling and also present the details of the problem of computing the amount of refrigeration necessary to accomplish the task of removing the required number of heat units.

Methods of Pre-cooling

Pre-cooling, as it usually is practiced, is accomplished in one of two general ways: (1) the warehouse system, by which method the fruit is graded and packed for shipment but stored in a refrigerated room for a short period before loading into the cars, or (2) the car pre-cooling method, in which the packed fruit is loaded directly into the car and there cooled before shipment by the circulation of chilled air through the car, about the load.

Warehouse System: With the warehouse system the produce generally is stacked in a refrigerated room and cooled to a temperature of 30-35 F before loading into the refrigerator cars or holds of ships. Cooling is accomplished by circulating cold air at a temperature of 28-30 F about the fruit, usually bringing in the cold air from ducts at one end of the room and removing it through ports at the opposite end, providing the necessary baffle plates to direct the flow of air through the produce to be cooled. The stacking of packages to present the maximum opportunity for contact with the circulating air is very important. The goods generally are stripped so that air movement about all sides of the packages is possible.

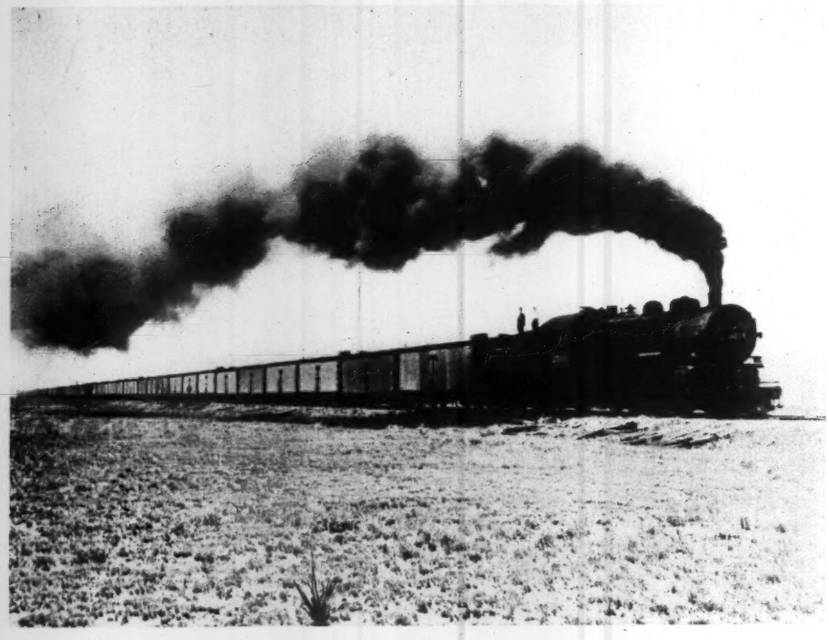
A large volume of air is necessary to remove sufficient heat from the packed fruit to accomplish the necessary temperature reduction on the inside of tight packages. Whitehouse (*7*) reports that 30 hours at 32 F were required to lower the temperature at the center of a box of wrapped apples from 70 to 40 F and an additional 130 hours to reduce the temperature to 34 F. A more open type of package would, of course have a more rapid rate of cooling. Air temperatures lower than 28 F are not advisable due to the possibility of freezing the fruit on the outside of the package or stack.

Car Pre-cooling: Car pre-cooling is accomplished in two ways: (1) by transporting the loaded car to a central cooling plant where by means of flexible ducts a stream of refrigerated air is passed through the ventilating system of the car, or, (2) by means of a portable car pre-cooling device which may be stationed alongside of the car at any convenient point. The first type is found in concentrated fruit districts on the west coast where heavy shipments are made from common shipping points. Some of these plants are equipped to pre-cool an entire train of cars simultaneously. The portable outfits are more adaptable to scattered or isolated shipping points or in the case of large growers who ship quantities of perishable produce. These portable car pre-coolers make use of the ice in the bunkers of the refrigerator car through which air is forced or drawn and hence are not so positive in action as systems using controlled artificial refrigeration.

Two types of portable car pre-coolers are described by Overholser and Moses (*8*). The first type consists of four ¾-hp. vertical electric motors with propeller fans mounted on a horizontal carriage which can be placed in the doorway space of the loaded car upon or between the center bracing. A canvas cover is placed over the top of the load in each end of the car from the ice bunker to the center bracing and fastened in place with horizontal struts. When the fans are in operation, cold air is drawn forward from the top of the ice bunkers, over the canvas and forced downward between the center bracings and sent backward through the air space between packages to the bottom of the ice bunkers. The warmed air passes through the ice, releases the absorbed heat and comes out again at the top of the bunker and is recirculated.

With this type of outfit the average air temperatures in different parts of the car did not drop below 45 F. The temperature changes within the cars were greatly influenced by the amount of ice in the bunkers and by the size of the channels formed between the cakes of melting ice.

As an example of what may be accomplished by this type of car pre-cooler these writers report that with a car of peaches in boxes the temperature of the fruit



100 Cars of Canteloupes from California Being Hauled by a Rock Island Freight Train

was reduced 15 F in nine hours while a similarly iced car without the pre-cooler reduced the temperature of the fruit less than 2 F during the same period. The refrigeration capacity of the latter car was almost entirely utilized in preventing a rise in temperature of the fruit from self-heating.

The second type of portable car pre-cooler described by these investigators consists of a large fan blower connected to one hatch by a collapsible canvas air duct about 12 in. in diameter. The air from the intake side is forced downward through the ice bunker, out at the bottom, through the fruit to the bottom of the opposite bunker, up through the ice to the outlet hatch, where another collapsible canvas duct leads to the intake of the blower. The upper openings of the bulkheads at each end of the car are covered with canvas to prevent the short circuiting of the air. By this method of pre-cooling the average drop in temperature of a car of tomatoes was 11 F in five hours. With a car of canteloupes to which 150 pounds of coarse salt was added to the 7,000 lb. of ice in the bunkers the temperature of the produce was reduced 20 F in six hours.

After a car is pre-cooled in this manner it should be re-iced at once and immediately closed to exclude warm air. A temperature rise will follow in most cases, the extent of the rise depending upon the amount of heat still retained by the fruit. In any case, however, the fruit treated in this manner has a decided advantage over fruit that has not been pre-cooled.

A simple type of car pre-cooler has been developed by Galloway (*9*) which consists of two ½-hp. motor-driven blowers of 1,000-c.f.m. capacity each. One blower is placed in each ice bunker directly under the bunker hatches with the discharge duct of the blower pointing through the top bunker openings into the body of the car. The upper bunker openings, with the exception of the space occupied by the blower discharge orifice, are covered with heavy paper so that the air movement may be controlled. To compensate for differences in bunker widths, adjustable sleeves are used from the blower discharge openings so that the opening of the discharge duct may fit closely against the bunker screen. The electric lead cords are brought out through the cushioned edges of the hatch plugs.

Each assembly is suspended under the hatch openings by flat adjustable web straps which are fastened by hooks at the upper ends to the running board on the top of the car on one side and under the eaves on the other side. This method of suspending the blower assembly permits the replacement of the hatch plugs during operation, eliminating air leaks. The length of the straps may be adjusted to suit different types of cars.

In the operation of this system a reversal of the natural air movement takes place. The circulation in each half of the car is upward through the ice, out over the top of the load and down through it, and back under the false floor into the lower part of the ice bunkers. Salt may be used with this system to secure a lower temperature. The paper should, of course, be removed from the bulkhead openings as soon as the pre-cooling process is completed.

Portable pre-cooling devices using the ice of the car as their source of refrigeration are limited to the temperature that may be secured from the melting ice or the ice and salt mixture. The warehouse method of pre-cooling or the utilization of mechanical refrigeration for this purpose will therefore give more positive means of rapid pre-cooling.

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Heating·Piping
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American Society of Heating and Ventilating Engineers

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JANUARY 1932

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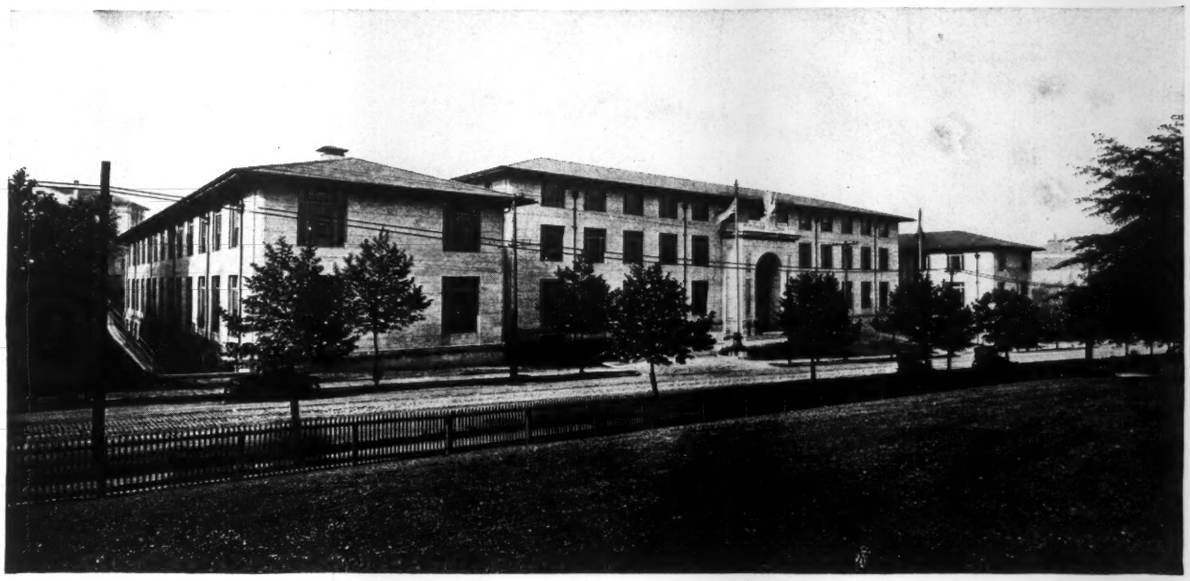

PITTSBURGH EXPERIMENT STATION OF THE U. S. BUREAU OF MINES WHERE THE RESEARCH LABORATORY OF THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS IS LOCATED

Heat Emission from Iron and Copper Pipe

By F. C. Houghten[1] (*MEMBER*) and Carl Gutberlet[2] (*NON-MEMBER*)
Pittsburgh, Pa.

MANY studies have been directed towards the determination of heat emission from bare and covered standard iron and steel pipe carrying steam and water at various temperatures. This paper presents data on the heat emission from bare copper pipe under service conditions, resulting from a study made at the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, in co-operation with the *Associated Copper Tubing Manufacturers*. A few data are included giving heat emission for galvanized, brass and aluminum pipe, the effect of painting the pipe on the rate of heat emission, and the reduction in heat emission from standard iron pipe resulting from the application of commercial insulation.

Since heat emission from iron pipe has been the subject of much research and analysis, and since the present study on copper tubing was of necessity brief in comparison, the study of heat emission from copper tubing was made comparative with heat emission from iron.

Test Set-Up and Procedure

All tests were made in the psychrometric chambers of the Laboratory in the Pittsburgh Experiment Station of the United States Bureau of Mines where surrounding atmospheric conditions could be accurately controlled. This room is located in the interior of the building with well insulated walls.

The inside wall surfaces in view of the test pipe were therefore at the same temperature as the surrounding air.

Fig. 1 is a photograph of the set-up used for studying horizontal pipe. Figs. 2 and 3 are drawings of the arrangement of the apparatus for studying vertical and horizontal pipe respectively. Steam, at a constant controlled pressure, held well within the limits of 0.1 in. of water column, entered the steam separator, A, at B, and passed through the electric superheater, C, to the two test pipes, D and D'.

The condensate was drained from the test pipe at E and E' through the water sealed traps, F and F', and was collected in graduated cylinders, G and G'. The tops of the graduate cylinders were covered so as to lessen possible re-evaporation from the receiver. A small thread of steam was allowed to escape at H and H' throughout the test in order to keep the system free from air.

Before starting a test the entire system was purged of air, and heated for a sufficient length of time to insure

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For presentation at the 38th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Cleveland, Ohio, January, 1932.

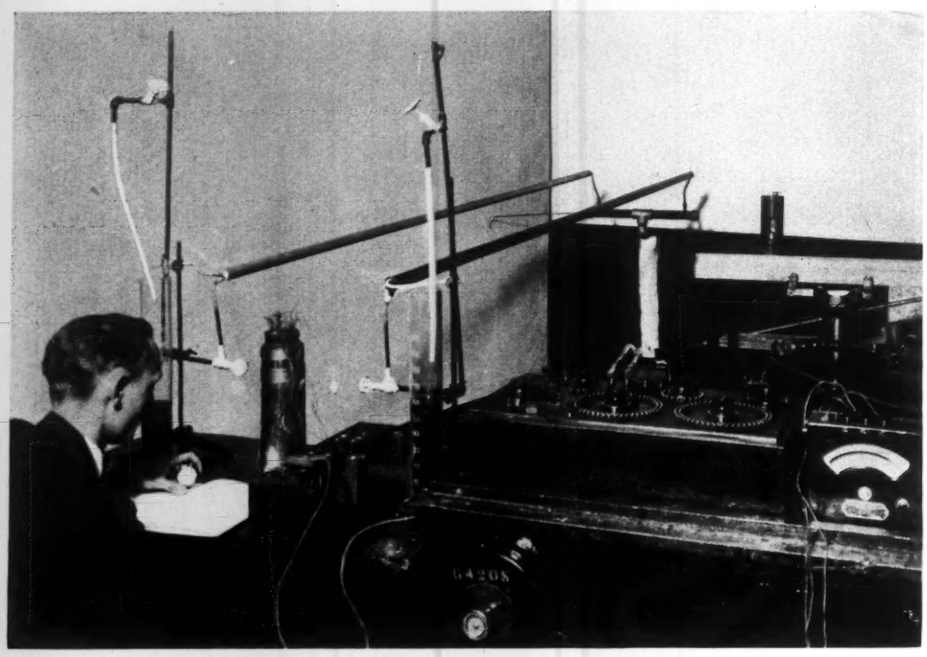

FIG. 1 — LABORATORY SET-UP FOR DETERMINING HEAT EMISSION FROM HORIZONTAL PIPE

equilibrium conditions. The valves, I and I', were adjusted to give equal steam pressure in both pipes. The electric current through the superheater was regulated to maintain approximately 0.1 deg of superheat in the entering steam as indicated by thermocouples located at J and K and K'.

Throughout the test, temperatures were observed at points K, L, M, N, and O. Thermocouples K and O, were located in ⅛-in. O.D. copper tubes, with the junctions in the steam inlet to, and the drain from, the pipe. The outside surface temperatures of the test pipes were observed at L, M, and N by thermocouples whose junctions were soldered into holes drilled part way through the walls of the test pipes, care being taken to allow the solder to cover as little of the pipe surface as possible. All thermocouples were of No. 40 B & S gage wire.

During the test, the water level in the water-sealed trap was maintained at a level 0.5-in. below the top of the glass section P of the trap to reduce the steam heated surface other than the pipe itself to a minimum.

The volume and temperature of the water in the receivers were observed and plotted every ten minutes, and all tests were continued until two consecutive periods of one hour showed the same condensation, or until the curve resulting from the 10-min observations gave a continuous straight line. The volumetric determination of condensation was checked by weighing at the beginning and end of each test.

The ends of the pipe were closed by discs soldered in place. The steam supply drain and air vent connections were of small copper tubing soldered in place. Samples of pipe 9 ft in length were studied. In the vertical position, the pipes reached from 8-in. above the floor to 6 in. below the ceiling, and were separated from each other by 30 in., and from the nearest wall by 30 in. The pipes studied in the horizontal position were 30 in. apart, pitched 2 in. in the 9-ft length, and were supported by fine wires. In the horizontal position, they were located 24 in. or more from any wall, and 48 in. from the floor. The necessary instruments, other equipment and the observer were located so as to offer no interference to normal convection currents.

Data and Results

All tests were made with steam temperatures ranging from 212.3 F to 213.5 F. The entering steam temperature measured at D and D', Figs. 2 and 3, was always from 0.3 to 0.5 deg higher than the temperature observed at the drains O and O'. Since the entering steam con-

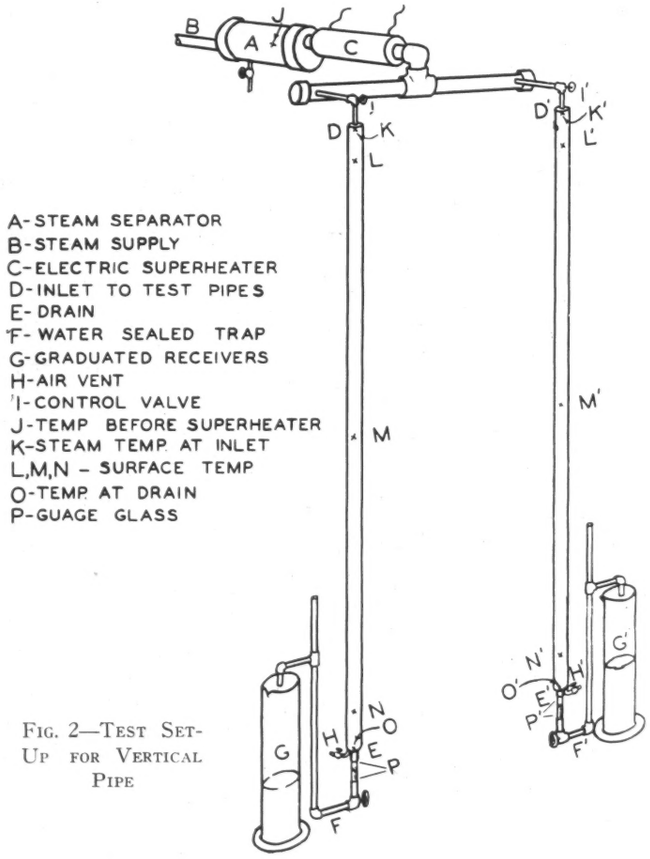

FIG. 2—TEST SET-UP FOR VERTICAL PIPE

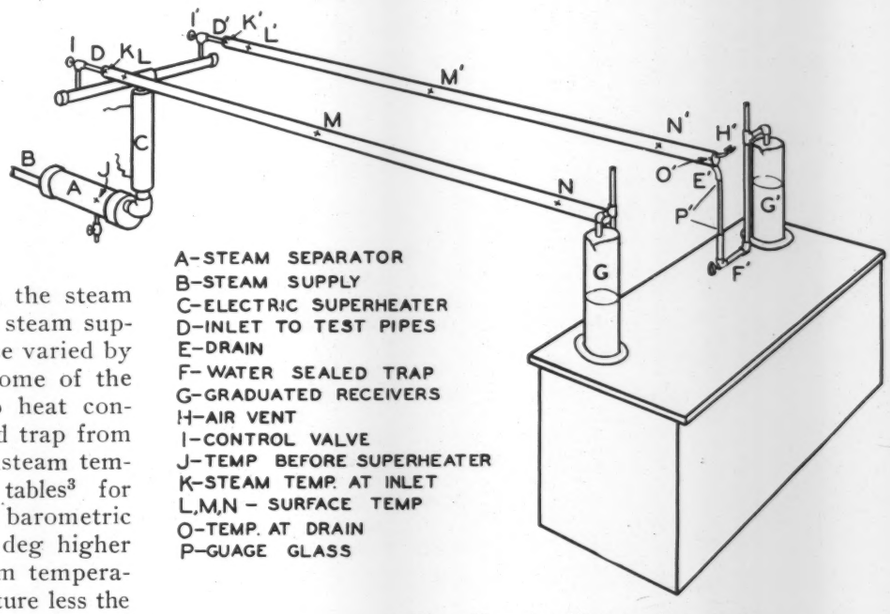

FIG. 3—TEST SET-UP FOR HORIZONTAL PIPE

tained about 0.1 deg superheat, the steam temperature as measured at the steam supply and the drain ends of the pipe varied by less than 0.4 deg. No doubt some of the observed difference was due to heat conduction through the water sealed trap from the lower end of the pipe. The steam temperature as taken from steam tables[3] for the observed steam pressure and barometric pressure was always about 0.3 deg higher than the observed entering steam temperature. The entering steam temperature less the degree of superheat was accepted as the steam temperature for heat loss computations.

The three thermocouples attached to the outside surface of the pipe invariably showed a temperature from 0.3 deg to 0.5 deg lower than the accepted steam temperature. A part of this difference is necessarily due to the temperature drop through the pipe wall. Because of the conduction of heat away from the surface through the fine thermocouple wires, the true temperature drop through the wall is less than this observed difference. For the purpose of calculating heat loss coefficients, the pipe surface temperature was assumed to be the same as that of the steam.

The following two coefficients were calculated for the bare pipe:

1. The thermal transmittance coefficient (U) which is the heat emitted in Btu per hour per square foot of surface per degree temperature difference between steam and air. Since the steam temperature was practically that of the pipe surface, this coefficient is also the film transmittance coefficient for the outside surface of the pipe. The thermal transmittance coefficient (U) is given by the formula:

$$U = \frac{r\,W}{A\ (t_s - t_a)} \qquad (1)$$

2. The heat emission from the pipe (H) in Btu per hour per foot length of pipe per degree temperature difference between the steam and air.

$H = U \times$ (square feet of pipe surface per linear foot of pipe) (2)

The following two coefficient were calculated for the insulated pipe:

1. The film transmittance coefficient (h_1) which is the heat emitted in Btu per hour per square foot of insulation surface per degree temperature difference between the surface of the insulation and the air. It is given by the formula:

$$h_1 = \frac{r\,W - U \times \text{(auxiliary surface area)}}{A_1\ (t_1 - t_a)} \qquad (3)$$

2. The heat emission through the insulation (H_1), in Btu per hour per foot length of insulated pipe per degree temperature difference between the steam and air.

$$H_1 = \frac{r\,W - U \times \text{(auxiliary surface area)}}{A_1\ (t_s - t_a)} \times \text{(Square feet insulation surface per foot length of pipe)} \qquad (4)$$

where

U = transmittance cofficient steam to air expressed in Btu per hour per square foot of pipe surface per degree Fahrenheit temperature difference steam to air.

H = Btu emitted per hour per foot length of bare pipe per degree Fahrenheit temperature difference steam to air.

h_1 = film transmittance coefficient for the insulation surface in Btu per hour per square foot of insulation surface per degree temperature difference insulation surface to air.

H_1 = Btu emitted per hour per foot length of insulated pipe per degree temperature difference steam to air.

r = latent heat of steam at temperature t_s

W = weight of condensate in pounds per hour

t_s = steam temperature in degrees Fahrenheit

t_a = air temperature in degrees Fahrenheit

t_1 = insulation surface temperature in degrees Fahrenheit.

A = area of pipe surface, including the auxiliary surface.

A_1 = area of insulation surface, including ends of insulation.

Auxiliary surface area = surface emitting heat from the condensation weighed other than the sides of the pipe. This includes ends of the pipe and the part of the water sealed trap containing steam.

A small error exists in the coefficients as calculated, because of the surface area of the ends of the pipe, the part of the drain containing steam, and the ends of the insulation. The auxiliary surface for the bare pipe did not include the steam supply connection for the reason that it contained superheated steam; neither did it include the air vent as steam condensed in this connection was not weighed. This surface for the ¾-in., 1-in., and 1¼-in. bare pipe, constituted from 1.5 per cent to 1.8 per cent of the total bare surface of the pipe, and 1.9 per cent and 2.2 per cent of the 4-in. copper and iron pipe respectively. If this auxiliary surface emitted as much heat per unit area as the sides of the pipe there would be no error. It is probable that this surface emitted less heat per unit area than the test pipe, and therefore, the error due to the extra surface was considerably less than the percentage this surface is to

[3] Steam Tables and Mollier Diagram, by Joseph H. Keenan.

the surface of the entire pipe. The ends of the insulation constituted 1.2 per cent to 1.3 per cent of the side surface area. Since the heat loss per unit area of the ends is probably less than that of the sides, the resulting error in the coefficient should be considerably less than 1 per cent.

TABLE 1—HEAT EMISSION FROM BARE AND COVERED IRON PIPE

Position	Size (Inches)	Bare			Covered	
		A. S. H. V. E. Lab. Results		Heilman	A. S. H. V. E. Lab. Results	
		U	H	H	h_o	H_I
Horizontal	¾	2.84	0.788	0.765	1.68	0.244
Horizontal	1	2.72	0.934	0.924	2.52	0.271
Horizontal	1¼	2.63	1.13	1.16	1.74	0.205
Horizontal	4	2.36	2.78	2.79	.....	.....
Vertical	¾	2.71	0.749	.....	1.88	0.240
Vertical	1	2.63	0.903	.....	2.00	0.251
Vertical	1¼	2.55	1.10	.....	2.00	0.306

U = Btu per hour per square foot per degree Fahrenheit steam to air; for bare pipe.
H = Btu per hour per foot length per degree Fahrenheit steam to air; for bare pipe.
h_o = Btu per hour per square foot per degree Fahrenheit insulated surface to air; for covered pipe.
H_I = Btu per hour per foot length per degree Fahrenheit steam to air; for covered pipe.

TABLE 2—HEAT EMISSION FOR BARE COPPER PIPE, BASED ON A. S. H. V. E. LABORATORY RESULTS

Position	Size (Inches)	U	H
Horizontal	¾	2.01	0.462
Horizontal	1	1.91	0.560
Horizontal	1¼	1.76	0.624
Horizontal	4	1.32	1.43
Vertical	¾	2.02	0.463
Vertical	1	1.92	0.574

U = Btu per hour per square foot per degree Fahrenheit steam to air; for bare pipe.
H = Btu per hour per foot length per degree Fahrenheit steam to air; for bare pipe.

TABLE 3—HEAT EMISSION FROM ONE-INCH HORIZONTAL PIPE WITH VARIOUS SURFACES IN BTU PER HOUR PER SQUARE FOOT OF SURFACE PER DEGREE FAHRENHEIT DIFFERENCE BETWEEN STEAM AND AIR

Type of Surface	Iron	Copper	Gal. Iron	Brass	Aluminum
Bare as received	2.72	1.91	1.78	1.71	1.93
Buffed	1.85	1.74	.....	.....	.....
Black Duco	2.87	.....	.....	.....	2.88
Lamp Black	2.88	.....	.....	.....	.....
Red Duco	.....	2.89	.....	.....	.....
White Duco	2.88	.....	.....	.....	.....
Aluminum Paint	2.24	.....	.....	.....	.....
Black Duco over Aluminum Paint	2.81	.....	.....	.....	.....
Aluminum Paint over Black Duco	.....	.....	.....	.....	2.19
Alum. Paint over Blk. Duco over Alum. Paint	2.17	.....	.....	.....	.....

Table 1 gives the heat emission in Btu per hour per square foot of pipe surface per degree temperature difference between steam and air, and the Btu per hour per foot length of pipe per degree temperature difference between steam and air, as determined by the A. S. H. V. E. Laboratory for the bare iron pipe. The table also gives the heat emission for bare-iron pipe as reported by Heilman.[4] The last two columns of the table give the surface transfer coefficients for the insulation surface in Btu per hour per square foot of insulation surface per degree temperature difference between the insulation surface and the air, and the heat emission in Btu per hour per foot length of insulated pipe per degree temperature difference between steam and air. Table 2 gives similar heat emission data for bare copper pipe in the vertical and horizontal positions. Table 3 gives the transmittance coefficients for bare, galvanized, aluminum and brass pipe, and for iron and copper pipe with other than *as received* surfaces. These results are for one-inch pipe only.

Discussion of Results

The small temperature difference existing between the steam and the outside surface of the pipe bears out the generally accepted fact that the important factor in resistance to heat flow from a bare pipe is in the transfer from the metal surface to the air, rather than the thermal resistance of metal itself.

The heat transfer coefficients expressed in Btu per hour per foot length of pipe per degree temperature difference steam to air, are plotted in Fig. 4 for bare iron, bare copper, and for insulated iron. When plotted against outside pipe diameter, substantially straight lines result. The bare iron curve fits almost perfectly the results of Heilman.[4]

For pipe of 1.65 in., outside diameter, which is the outside diameter of 1¼-in. nominal size iron pipe, the bare iron, bare copper, and covered iron, emit 1.13, 0.70, and 0.31 Btu per foot of pipe respectively. If the comparison is made on the basis of 1¼-in. nominal size, the bare iron, bare copper, and covered iron emit 1.13, 0.62, and 0.31 Btu per foot of pipe respectively, or the 1¼-in. copper tube emits 54 per cent as much heat as the 1¼-in. bare iron, and 200 per cent as much as the covered iron.

The data show little variation in heat emission from vertical and horizontal pipe. The greatest variation is shown for the bare iron, in which case the vertical pipe shows a lower rate of heat emission. This is contrary to what was expected. Fig. 5 shows the relationship between the heat emitted in Btu per hour per square foot of pipe surface per degree temperature difference steam to air. These curves show a decrease in the rate of transfer per unit area with pipe size.

The heat transfer values for galvanized, brass and aluminum pipe, and for the iron and copper pipe with surfaces other than *as received*, show higher rates of heat emission for pipe painted with any kind of pigment paint, than for the same pipe bare. The black Duco, lamp black, red Duco, and white Duco when used on any pipe gave values ranging from 2.87 to 2.89, excepting in the case of the black Duco over aluminum

[4] Heat Transmission from Bare and Insulated Pipes, by R. H. Heilman (Industrial and Engineering Chemistry, May, 1924).

paint, which gives a value of 2.81. A value of 2.88 can therefore be accepted for surfaces painted with pigment paint. This value is higher than that found by Willard and Kratz [5] for a 10-in. painted cylinder. The curves in Fig. 5 show a decrease in the coefficient with increase in size of the cylinder, which probably accounts for the difference. The value of 2.72 for bare iron as received seems low when compared with 2.88 for the same surface painted. Upon examination of the pipe as received, it was observed that while the greater part of the surface was black, many small areas were bright metal.

A comparison of the value of 1.93 for bare aluminum and the values of 2.17 to 2.24 for aluminum painted surfaces, whether applied directly to the metal or over other paints, shows that aluminum paint has an emissivity not much greater than bare aluminum. A comparison of the values for copper as received, copper buffed, galvanized, brass and aluminum shows that 1-in. pipe of any of these metallic surfaces give values ranging from 1.71 to 1.93. The galvanized and brass pipe were washed with gasoline to remove oil and grease deposits found on the pipe as received.

[5] Heat Emission from the Surfaces of Cast Iron and Copper Cylinders Heated with Low Pressure Steam, by A. C. Willard and A. P. Kratz (*Heating, Piping and Air Conditioning*, Feb., 1931).

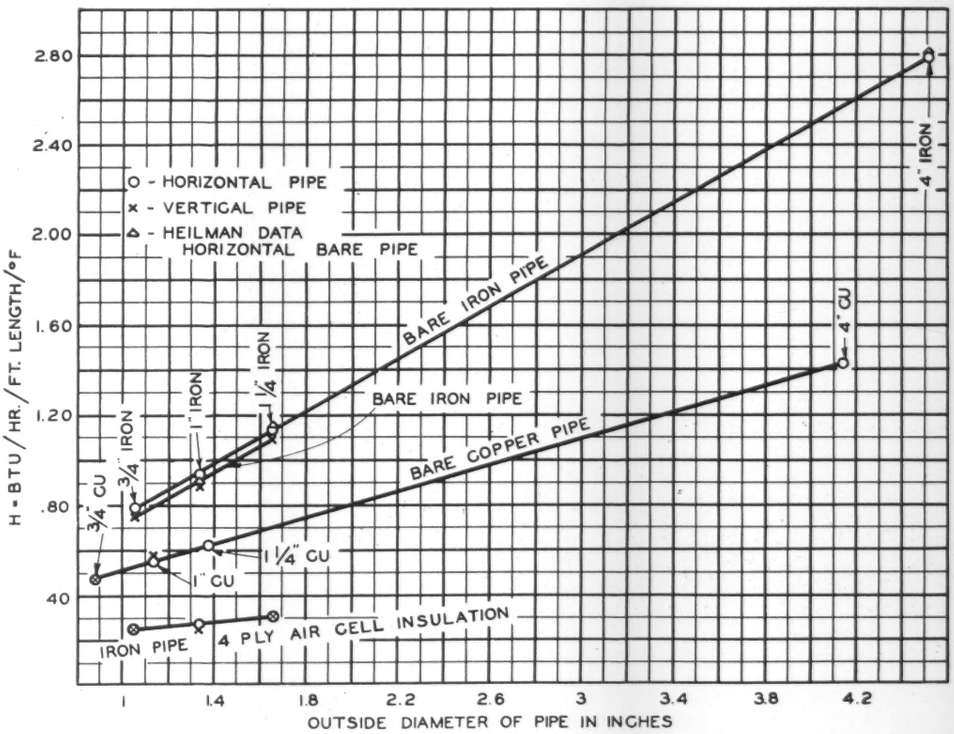

Fig. 4—Relation Between Heat Emission per Foot of Length and Diameter of Pipe

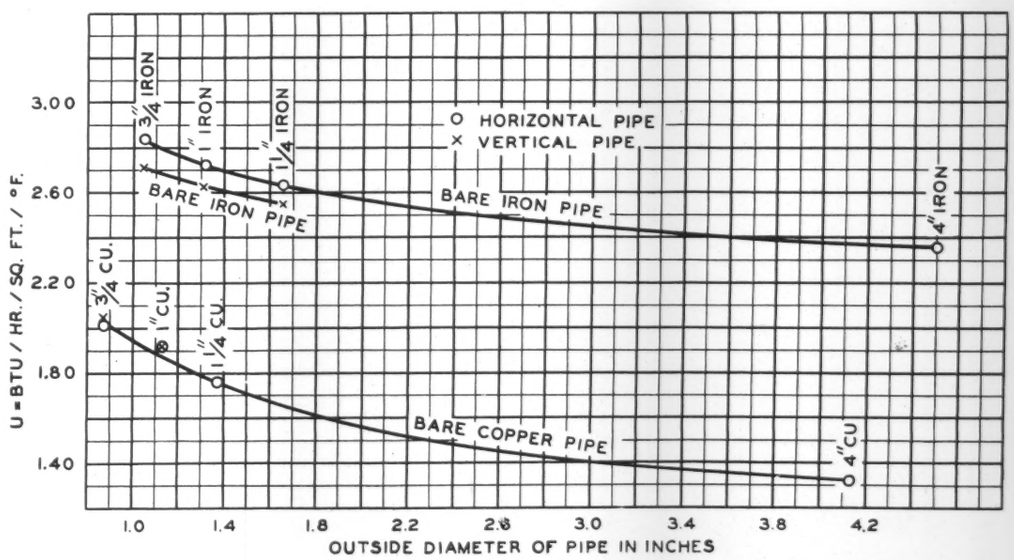

Fig. 5—Relation Between Heat Emission per Sq. Ft. of Surface and Pipe Diameter

It is of interest to compare the heat loss from the pipe as determined by test with values calculated from the Stefan-Boltzman radiation equation and the Rice[6]-Heilman[7] convection equation. The Stefan-Boltzman equation is:

$$H_r = 0.1723\, e \left[\left(\frac{T_1}{100} \right)^4 - \left(\frac{T_2}{100} \right)^4 \right]$$

where

H_r = heat loss by radiation, Btu per square foot per hour
e = emissivity coefficient
T_1 = absolute temperature of the hot surface, degrees Fahrenheit
T_2 = absolute temperature of the surroundings, degrees Fahrenheit

Values for e which might be chosen for the copper and iron surfaces studied are given in Table 4.

TABLE 4—VALUE OF e FOR COPPER AND IRON SURFACES

SURFACE	e	AUTHORITY
Wrought Iron high polish	0.28	Marks Handbook[a]
Cast Iron—bright	0.21	International Critical Tables
Rough Steel	0.94	Heilman[b]
Copper—buffed high polish	0.10	Marks handbook
Copper—buffed	0.167	Hütte[c]
Copper—oxidized	0.57	International Critical Tables

[a] Transmission of Heat by Radiation, Revised by H. C. Hottel. (Marks Handbook.)
[b] Surface Heat Transmission, by R. H. Heilman. (*Mechanical Engineering*, Section 1, May, 1929.)
[c] Des Ingenieurs Taschenbuch, Twenty-second edition.

The Rice-Heilman equation is:

$$H_c = C \left(\frac{1}{D} \right)^{0.2} \times \left(\frac{1}{T_a} \right)^{0.181} \times \Delta t^{1.266}$$

where

H_c = heat loss by convection, Btu per square foot per hour
C = a constant depending on configuration. 1.016 for horizontal cylinders
D = diameter, inches
T_a = average of the absolute temperatures of the surface and the surrounding air, degrees Fahrenheit
Δt = temperature difference between the surface and the surrounding air, degrees Fahrenheit

Table 5 gives the heat loss in Btu per hour per square foot per degree temperature difference for copper and iron pipe of different surfaces as found by the Laboratory and as calculated by the Stefan-Boltzman equation, the Rice-Heilman equation, and assumed values for e.

TABLE 5—HEAT LOSS FROM ONE-INCH LOW PRESSURE STEAM PIPE AS FOUND BY TEST AND BY CALCULATION

KIND OF PIPE	ASSUMED VALUE OF e	H_r	H_c	$\frac{H_r+H_c}{\Delta t}$	A.S.H.V.E LABORATORY
Copper as received	0.57	122.97	215.59	2.38	1.91
Copper—buffed	0.167	35.60	210.05	1.76	1.74
Copper—buffed	0.10	22.81	210.05	1.67	1.74
Iron—black	0.94	198.67	200.28	2.89	2.72
Iron—buffed	0.21	45.43	210.49	1.79	1.85
Iron—buffed	0.28	60.57	210.49	1.90	1.85
Any pipe covered with pigment paint	0.94	198.67	200.28	2.89	2.87 to 2.89

H_r = Heat emission by radiation in Btu per hour per square foot of surface for the existing temperature difference.
H_c = Heat emission by convection in Btu per hour per square foot of surface for the existing temperature difference.
$\frac{H_r + H_c}{\Delta t}$ = Total heat emission in Btu per hour per square foot of surface per degree temperature difference.

[6] Chester W. Rice (*A.I.E.E.* Transactions, Vol. 43, 1924).
[7] Surface Heat Transmission, by R. H. Heilman (*Mechanical Engineering*, May, 1929, Section 1).

The calculated value for the copper pipe based upon $e = 0.167$ gives 1.76 Btu compared with 1.74 found by the A. S. H. V. E. Laboratory for a buffed pipe, and 1.91 found for the copper pipe as received. Obviously the value $e = 0.10$ for a highly polished copper and the value of 0.57 for oxidized copper are too low and too high respectively for the buffed and as received copper pipe studied by the A. S. H. V. E. Laboratory.

The values of $e = 0.21$ and 0.28 for bright cast iron and polished wrought iron giving calculated values of 1.71 and 1.90, are respectively too low and too high for the buffed iron pipe which Laboratory test found to have an emission of 1.85 Btu per hour per square foot per degree temperature difference. The value of $e = 0.94$ giving a calculated coefficient of 2.89 is too high for the black iron, which by test gave a coefficient of 2.72, but it is about right for the pigment painted surfaces tested by the Laboratory, which gave coefficients ranging from 2.87 to 2.89 Btu per hour per square foot per degree temperature difference. It is of interest to note that the calculations show that for a 1-in. low pressure steam pipe with a surface having an emissivity e of approximately 100, the radiation and convection losses each take care of about one-half of the total heat loss. Thus for an emissivity of $e = 0.94$, radiation and convection account for 198.7 and 200 Btu respectively.

These comparisons show that with the selection of the proper value for e the rate of heat emission for any case in hand may be accurately computed. However, the selection of the value of e for any particular case is often difficult.

Summary and Conclusions

1. The heat loss from bare copper pipe is approximately 54 per cent of the loss from bare black iron pipe of the same nominal size, and 203 per cent of the loss from iron pipe covered with four-ply air cell insulation.
2. The heat loss is approximately the same for horizontal and vertical pipe of the same size and material.
3. The heat loss from pipe may be calculated from the Stefan-Boltzman and the Rice-Heilman equations with a high degree of accuracy by using the proper shape factor and value of e. However, the choice of e for the case in hand is important.

VALUES OF e FOR VARIOUS SURFACES[a]

SURFACE	TEMPERATURE, DEGREES FAHRENHEIT				
	100	200	300	400	500
Polished silver	0.0221	0.0252	0.0292	0.0315	0.0295
Lampblack	0.945	0.945	0.945	0.945	0.945
Asbestos Paper	0.930	0.934	0.943	0.955	0.929
Rough Steel Plate	0.945	0.950	0.955	0.961	0.969
Aluminum-Surfaced Roofing	0.216	.....	.....	.....	.....
Polished Brass	0.096	0.096	0.098	0.098	0.096
Flat Black Lacquer	0.96	0.98	.....	.....	.....
Black Lacquer	0.80	0.95	.....	.....	.....
White Lacquer	0.80	0.95	.....	.....	.....

[a] A. S. H. V. E. GUIDE 1931

Supplementary Friction Heads in One-Inch Cast-Iron Tees

By F. E. Giesecke[1] (*MEMBER*) and W. H. Badgett[2] (*NON-MEMBER*)
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This paper is the result of research conducted at Agricultural and Mechanical College of Texas in cooperation with the A. S. H. V. E. Research Laboratory.

IN THE first report of this study, which was presented at the Semi-Annual Meeting of the Society, Swampscott, Mass., June, 1931[3], it was shown that the loss of head in a tee varies with the direction in which the water flows through the tee and also with the proportion of the water which is diverted at right angles in the tee.

It was shown, for example (see Figs. 1 and 2) that

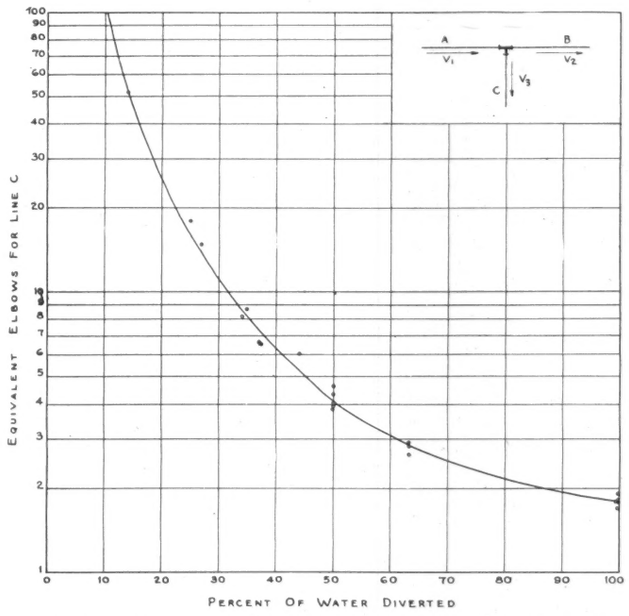

FIG. 1—THE FRICTION HEAD IN A 1 X 1 X 1-IN. CAST-IRON TEE WHEN WATER ENTERS AT *A* AND A PORTION IS DISCHARGED AT *C*, IN TERMS OF THE FRICTION HEAD OF AN ELBOW AT *C*

when the water enters at one end of the tee and 50 per cent is diverted while 50 per cent flows straight through the tee, the loss of head for the diverted portion is equal to that in four elbows and the loss of head for the remaining 50 per cent, which flows straight through the tee, is equal to that in 0.6 elbows. In expressing the loss of head in a tee in terms of the loss of head in an elbow, *i.e.*, in elbow equivalents, the loss of head in one elbow was assumed equal to $0.9 \frac{v^2}{2g}$, because in an earlier investigation, the loss of head in a standard 1-in. elbow had been found equal to this quantity. In more recent investigations by A. P. Kratz, H. J. Macintire, and R. E. Gould at the University of Illinois[4], it was found that for 1¼-in. and 2-in. elbows the losses of head varied from $0.89 \frac{v^2}{2g}$ to $1.33 \frac{v^2}{2g}$, averaging $1.035 \frac{v^2}{2g}$ for 104 tests. It will therefore be sufficiently accurate for practical purposes to place the loss of head in one elbow equal to $\frac{v^2}{2g}$. This will be done in the following discussions.

Loss of Head for Straight Flow

Before the tests of the losses of head in tees had been performed, it was believed that the loss of head for the portion of water flowing straight through the tee would be practically zero and might even be negative because

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[2] Research Assistant, Texas Engineering Experiment Station, Agricultural and Mechanical College of Texas, College Station, Texas.
[3] See Friction Heads in One-Inch Standard Cast-Iron Tees, by F. E. Giesecke and W. H. Badgett (*Heating, Piping and Air Conditioning*, May 1931, pp. 430 to 432).
[4] See University of Illinois Engineering Experiment Station *Bulletin* No. 222, pp. 21 and 22.

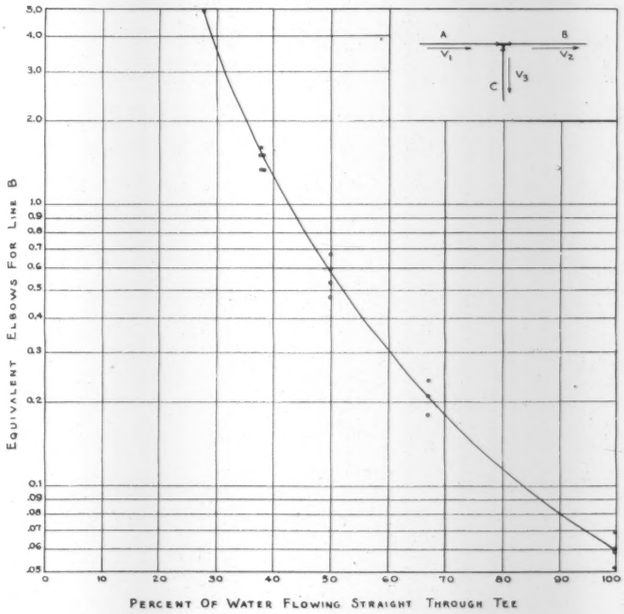


FIG. 2—THE FRICTION HEAD IN A 1 X 1 X 1-IN. CAST-IRON TEE WHEN WATER ENTERS AT *A* AND A PORTION IS DISCHARGED AT *B*, IN TERMS OF THE FRICTION HEAD OF AN ELBOW AT *B*

when a portion of the water is diverted the velocity of the remainder is reduced and it was believed that the resulting reduction of velocity head would be as large and possibly larger than the friction head in the tee.

Since the tests proved conclusively that this was not the case, it was decided to substitute a glass tee for the cast-iron tee and to observe the flow of water through the glass tee. In order to make the stream lines visible, small bubbles of gas (hydrogen and oxygen) were produced electrolytically in the fluid stream by a method similar to that described by J. Zenneck[5]. The resulting streams of gas bubbles could be seen very clearly and could be photographed when they were illuminated by a thin sheet of light at right angles to the direction from which the photograph was taken. Fig. 3 shows the stream lines and eddies in the glass tee when about 50 per cent of the water was being diverted. It is evident from the picture that no portion of the water flows straight through the tee, but that the entire current is deflected to such an extent that considerable eddying occurs also in the straight portion of the tee. This eddying is, no doubt, the cause of the unexpected loss of head in that portion of the current which passes straight through the tee and which was disclosed by the recent tests.

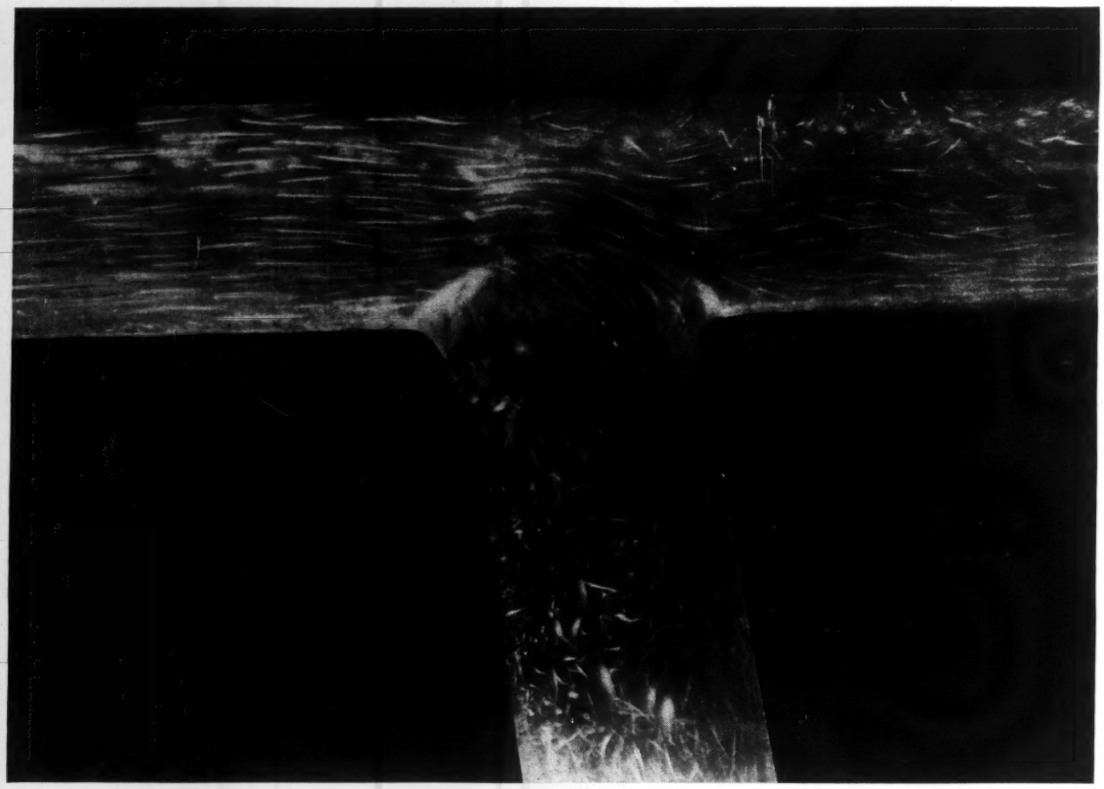


FIG. 3—THE STREAM LINES AND EDDIES OF WATER FLOWING THROUGH A TEE WHEN ABOUT 50 PER CENT OF THE WATER IS DIVERTED

Tests for Variations in Flow Direction

Having determined the losses of head when the water enters the tee at A and is discharged at B and C (Figs. 1 and 2), additional tests were made to determine the losses of head when the water enters at A and C and is discharged at B (Figs. 4 and 5); when it enters at C and is discharged at A and B (Fig. 6); and when it enters at A and B and is discharged at C (Fig. 7). The diagrams of these figures are to be used as follows:

1. In Fig. 4, if 25 per cent of the water flowing through the tee enters from C with the velocity v_3, the loss of head in the tee, for the path from C to B is equal to 10 elbow equivalents of the line C, *i. e.*, equal to $10 \frac{v_3^2}{2g}$.

2. In Fig. 5, if 75 per cent of the water flowing through the tee enters from A with the velocity v_1, the loss of head in the tee for the path AB is equal to 1.25 elbow equivalents of the line A, *i. e.*, equal to $1.25 \frac{v_1^2}{2g}$. It is interesting to note how much larger the loss of head is in the case of Fig. 4 than in the case of Fig. 2 for the water flowing straight through the tee; with 50 per cent flowing straight through, the two losses of head are 2.25 and 0.6 elbow equivalents, respectively.

3. In Fig. 6, if 25 per cent of the water flowing through the tee is discharged at A (or B) the loss of head in the tee for the path CA (or CB) is equal to 11 elbow equivalents for the line A (or B), *i. e.*, equal to $11 \frac{v_1^2}{2g}$ (or $11 \frac{v_2^2}{2g}$).

4. In Fig. 7, if 25 per cent of the water flowing through the tee enters at A, (or B) the loss of head in the tee for the path AC (or BC) is equal to 16 elbow equivalents for the line A (or B), *i. e.*, equal to $16 \frac{v_1^2}{2g}$ (or $16 \frac{v_2^2}{2g}$).

The diagrams of Figs. 1, 5, 6 and 7 are very similar. They are shown collectively by the four dashed light lines in Fig. 8. The differences between these four curves are so slight that they may be neglected for prac-

[5] Transactions, *German Association of Physicists*, Vol. XVI, No. 14.

tical purposes and the four curves replaced by one curve as shown by the heavy line of Fig. 8.

It should be noted that as the water, when diverted to C (Fig. 1), approaches the tee from A with the velocity v_1, the velocity in the direction AB is changed from v_1 to zero, and that the corresponding velocity head $\left(\frac{v_1^2}{2g}\right)$ must be transformed partly into heat and partly into pressure head; also, as the water flows from the tee to C its velocity increases from zero to v_3 and

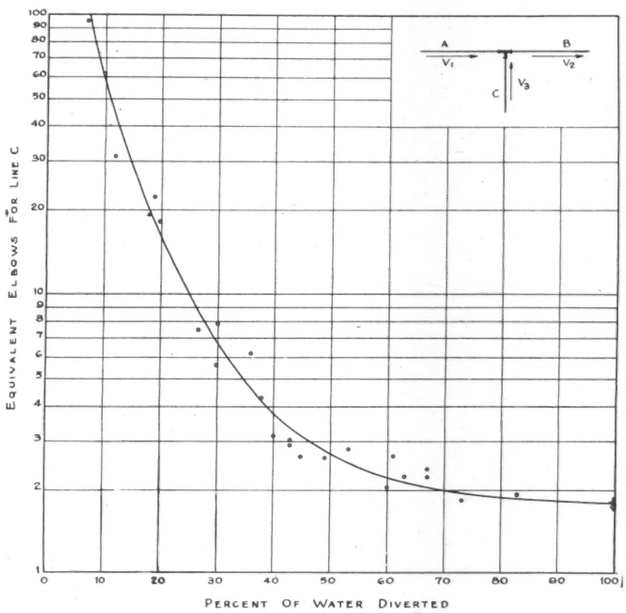


FIG. 4—THE FRICTION HEAD IN A 1 X 1 X 1-IN. CAST-IRON TEE WHEN WATER ENTERS AT A AND C AND IS DISCHARGED AT B, IN TERMS OF THE FRICTION HEAD OF AN ELBOW AT C

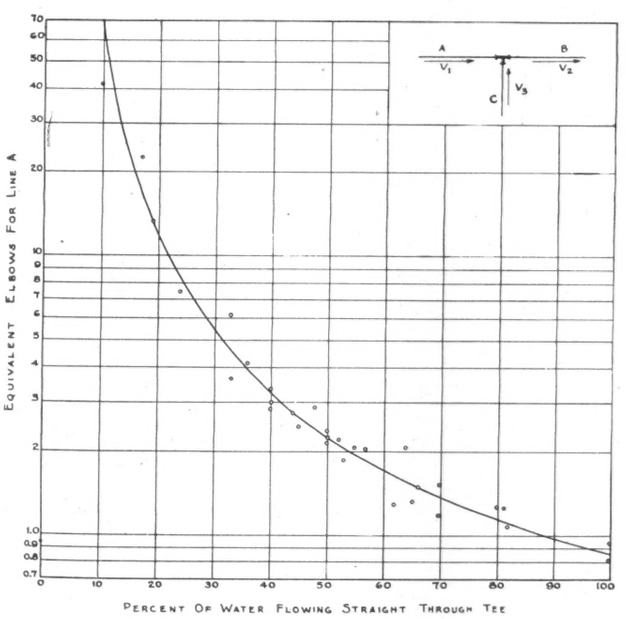


FIG. 5—THE FRICTION HEAD IN A 1 X 1 X 1-IN. CAST-IRON TEE WHEN WATER ENTERS AT A AND C AND IS DISCHARGED AT B, IN TERMS OF THE FRICTION HEAD OF AN ELBOW AT A

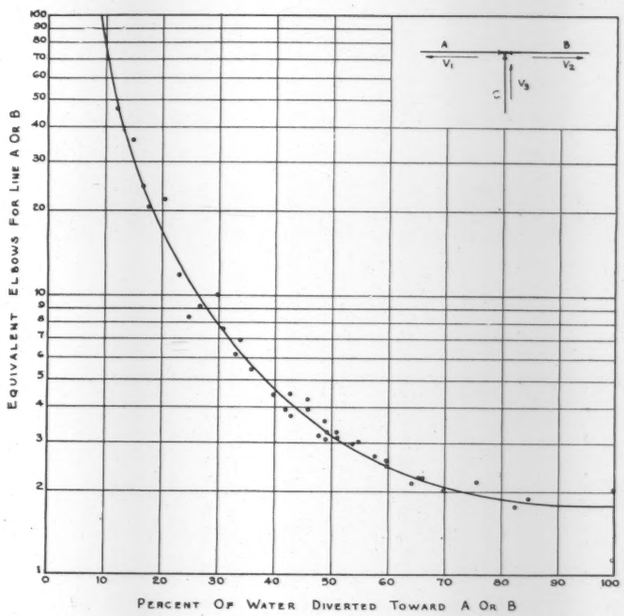


FIG. 6—THE FRICTION HEAD IN A 1 X 1 X 1-IN. CAST-IRON TEE WHEN WATER ENTERS AT C AND IS DISCHARGED AT A AND B, IN TERMS OF THE FRICTION HEAD OF AN ELBOW AT A OR B

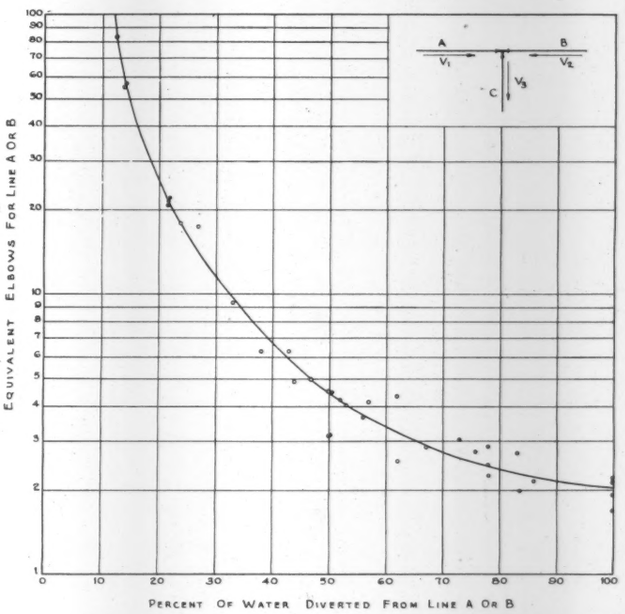


FIG. 7—THE FRICTION HEAD IN A 1 X 1 X 1-IN. CAST-IRON TEE WHEN WATER ENTERS AT A AND B AND IS DISCHARGED AT C, IN TERMS OF THE FRICTION HEAD OF AN ELBOW AT A OR B

that the corresponding velocity head $\left(\frac{v_3^2}{2g}\right)$ must be produced by a corresponding reduction in the pressure head. During one of the tests an effort was made to determine how much the pressure head builds up in the tee as the velocity of the water, in the direction from A to the tee, is reduced from v_1 to zero, and it was found that the increase in pressure head was too small to be measured by the methods used in the test. If it is assumed that the entire velocity head $\left(\frac{v_1^2}{2g}\right)$ is

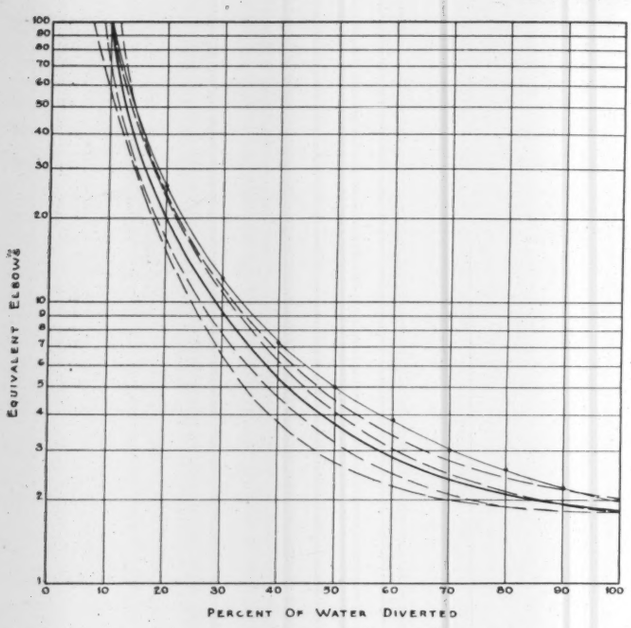


FIG. 8—THE FRICTION HEADS IN A 1 X 1 X 1-IN. CAST-IRON TEE AS SHOWN IN FIGS. 1, 5, 6 AND 7, WITH AN AVERAGE OF THESE FOUR LINES, AND A LINE SHOWING THEORETICAL VALUES CALCULATED BY AN APPROXIMATE METHOD

transformed into heat by the eddy currents in and near the tee, and if it is also assumed that the entire velocity head $\left(\frac{v_3^2}{2g}\right)$ is produced at the expense of the available pressure head, the total loss of head in the tee will be $\frac{v_1^2}{2g}$ plus $\frac{v_3^2}{2g}$.

Comparison of Calculated and Experimentally Determined Heads

To compare the losses of head calculated in this manner with those found experimentally and shown in Figs. 1, 5, 6 and 7, v_1 was expressed in terms of v_3 as follows:

If 50 per cent of the water is diverted, v_1 is equal to $2v_3$ and $\frac{v_1^2}{2g}$ is equal to $4\ \frac{v_3^2}{2g}$, and the total loss of head is equal to $5\ \frac{v_3^2}{2g}$ or to 5 elbow equivalents.

Similar calculations were made for other percentages and the results shown in Fig. 8 by small circles connected by a solid line. It will be noticed that this line practically coincides with the line shown in Fig. 7 for the case when the two currents from *A* and *B* impinge upon each other, but that it gives values about 20 per cent higher than the average of the four curves determined experimentally.

It seems from this comparison that the suggested method of calculating the losses of head in tees may be used with safety for all cases. For example, for a 1½ x 1¼ x 1-in. tee, placed as shown in Fig. 1, assume that 25 per cent of the water is diverted into the 1-in. pipe; v_1 will then be equal to v_3 multiplied by 4, and divided by 2.36, the ratio of the two pipe areas; *i.e.*, v_1 will be equal to $1.7\ v_3$. The corresponding loss of head in the tee will be $\frac{v_1^2}{2g}$ plus $\frac{v_3^2}{2g}$ or $3.9\ \frac{v_3^2}{2g}$, or 3.9 elbow equivalents for the line *C*.

For the losses of head in the currents passing straight through the tee, in the cases shown in Figs. 2 and 5, it is evident that the losses in the first case are so small that they may generally be neglected, but in the second case they are of sufficient magnitude to be considered in accurate calculations especially when the portion flowing straight through is from 25 to 75 per cent of the total.

Application of Results to Typical Case

A case which occurs frequently in practice and for which it is important to know the losses of head in tees is shown in Fig. 9. When the four radiators are of equal size, one-fourth of the water is diverted to the first floor radiator, one-third to the second floor radiator, and one-half to the third floor radiator. If the installation shown were a part of a forced circulation system

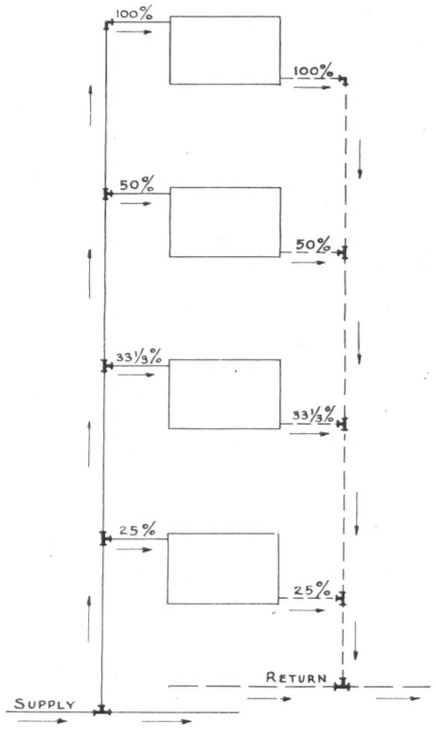


FIG. 9—A COMMON INSTALLATION IN WHICH THE FRICTION HEADS IN TEES ARE AN IMPORTANT PART OF THE TOTAL FRICTION HEADS IN THE SEVERAL CIRCUITS

using 1-in. pipe for the two risers and the four radiator connections, in an extreme case, the losses of head in the three pairs of tees would be, according to Figs. 1 and 4, for the first, second and third floor radiators, 26, 15 and 7 elbow equivalents respectively. As a result, the first floor radiator would be at a considerable disadvantage when compared with the second or third floor radiators. This disadvantage, resulting from lack of knowledge regarding the losses of head in tees, may be responsible in part for the frequently unsatisfactory operation of first floor radiators when connected as shown in Fig. 9.

Physical and Physiological Aspects of Air Conditioning

By C. P. Yaglou[1], Boston, Mass.
MEMBER

FROM the standpoint of comfort and health, air conditioning may be regarded as the art of maintaining the atmosphere of occupied spaces at a condition best suited to the physiologic requirements of the human body. According to the present knowledge of the problem, the requirements consist of maintaining simultaneously the proper temperature, humidity and air movement, and a reasonable air purity with regard to dusts, bacteria and odors.

Although the need for a pure air supply to occupied rooms is generally recognized, the reasons and specific requirements for optimum air conditions are not fully understood. By the process of elimination the oxygen, carbon dioxide, and crowd poison theories have long been discredited, while air-borne infections are now known to constitute but a very limited mode of transmission of disease in the immediate vicinity of infected persons.

For the past 20 years, temperature, humidity, air movement, and body odor have reigned supreme in the minds of physiologists and engineers, but the problem has not been completely solved. There seems to be a growing belief that indoor air, however treated mechanically, does not measure up to the same standard of open country air. According to the latest theory on the subject there is, apparently, a vital quality in outdoor air which is lost when the air is brought indoors, particularly when the air is conditioned mechanically. Ionization has been suggested, but has not yet been identified.[2]

Heat Regulation in Man

The importance of temperature, humidity and air movement arises from the profound influence which these factors exert upon body temperature, comfort and health. Body temperature is a resultant of two factors, (1) heat production and (2) heat loss. The heat resulting from the combustion of food within the body maintains its temperature well above that of the surrounding air. At the same time, heat is constantly lost from the lungs and skin by radiation, conduction and evaporation. Since, under ordinary conditions, the body temperature is maintained at its normal level of about 98.6 F, the heat production must be balanced by the heat loss. In healthy persons this takes place automatically by the action of the heat regulating mechanism.

According to the general view, special areas in the skin are sensitive to temperature. Nerve courses carry the sense impressions to the brain and the response comes back over another set of nerves, the motor nerves, to the musculature and to all the active tissues in the body, including the endocrine glands. In this way, a two-sided mechanism controls the body temperature by (1) regulation of internal heat production (chemical regulation), and (2) regulation of heat loss by means of automatic variation in the rate of cutaneous circulation and the operation of the sweat glands (physical regulation). The mechanisms of adjustment are complex and little understood at the present time. Coordination of these different mechanisms seems to vary greatly with different air conditions.

In reasonably warm environments (75 F to 80 F), metabolism, or internal heat production, is decreased to some extent, probably by an inhibitory action on heat producing organs, such as the liver and the spleen. The blood capillaries in the skin become dilated by reflex action of the vasomotor nerves, allowing more blood to flow into the skin, and thus increase its temperature and consequently its heat loss. The increase in peripheral circulation is at the expense of the internal organs. If this method of cooling is not in itself sufficient, the stimulus is extended to the sweat glands which allow water to pass through the surface of the skin, where it is evaporated. This method of cooling is the most effective of all, as long as the humidity of the air is sufficiently low to allow for evaporation. In high humidities, equally good results may be obtained by increasing the air movement, and hence heat loss by conduction and evaporation.

In cold environments, in order to keep the body warm there is an actual increase in metabolism brought about partly by voluntary muscular contractions (shivering) and partly by an involuntary reflex upon the heat producing organs. The surface blood vessels become constricted and shrink further below the surface, thus increasing the insulating layer and decreasing heat loss. The blood supply to the skin is curtailed by vasomotor shifts to the internal organs, in order to conserve body heat.

Influence of Adverse Air Conditions

Effects of Heat

Within certain limits the human organism is capable of adapting itself to changes in external temperature and humidity. The heat regulating center fails if the external temperature is so abnormally high that bodily heat cannot be eliminated as fast as it is produced. Part of it is retained in the body, causing a rise in skin and deep tissue temperature, an increase in the heart rate, ac-

[1] Assistant Professor in Industrial Hygiene, Harvard School of Public Health.

[2] Changes in Ionic Content in Occupied Rooms, Ventilated by Natural and Mechanical Methods, by C. P. Yaglou, L. C. Benjamin and S. P. Choate. (*Heating, Piping and Air Conditioning*, October, 1931).

TABLE 1—PHYSIOLOGICAL RESPONSES TO HEAT OF MEN AT REST AND AT WORK[a]

Effective Temp. (Deg Fahr)	Men at Rest				Men at Work 90,000 Foot—Pounds per Hour			
	Actual Cheek Temp. (Deg Fahr)	Rise in Rectal Temp. (Deg Fahr per hr)	Increase in Pulse Rate (Beats per Min per Hr)	Approximate Loss in Body Weight by Perspiration (Lb per Hr)	Total Work Accomplished (Ft Lb)	Rise in Body Temperature (Deg Fahr per Hr)	Increase in Pulse Rate (beats per Min per Hr)	Approximate Loss in Body Weight by Perspiration (Lb per Hr)
60					225,000	0.0	6	0.5
70		0.0	0	0.2	225,000	0.1	7	0.6
80	96.1	0.0	0	0.3	209,000	0.3	11	0.8
85	96.6	0.1	1	0.4	190,000	0.6	17	1.1
90	97.0	0.3	4	0.5	153,000	1.2	31	1.5
95	97.6	0.9	15	0.9	102,000	2.3	61	2.0
100	99.6	2.2	40	1.7	67,000	4.0	103[b]	2.7
105	104.7	4.0	83	2.7	49,000	6.0[b]	158[b]	3.5[b]
110		5.9[b]	137[b]	4.0[b]	37,000	8.5[b]	237[b]	4.4[b]

[a] Data by A. S. H. V. E. Research Laboratory.
[b] Computed value from exposures lasting less than one hour.

celerated respiration and the like (see Table 1). The metabolic rate increases also owing to the excessive rise in body temperature, and in extreme conditions a vicious cycle may result which eventually leads to serious physiologic damage.

Examples of this are met with in unusually hot summer weather and in hot industries where the radiant heat from hot objects renders heat loss from the body by radiation and convection impossible. Consequently, the workers depend entirely on evaporation for the elimination of body heat. They stream with perspiration and drink liquids abundantly to replace the loss.

Acute overheating leads to irritability and restlessness. The victims usually complain of headaches, dizziness, confusion, loss of appetite, indigestion, and distressing palpitations of the heart. The rise in body temperature and the vasomotor shifts of blood from the internal organs to the skin capillaries cause a diminution in the secretion of hydrochloric acid by the stomach and this leads to an alkaline condition favorable to the development of rich fecal bacteria in the intestinal tract.[3] The prevalence of gastro-intestinal disorders in hot weather may be partly or wholly explained by such changes in acid-base equilibrium. Continuous exposure to abnormal air conditions lowers the tone of the vital organs, and renders the system susceptible to disease, and to premature old age.

In warm atmospheres, particularly during physical work, a considerable amount of chloride is lost from the system through sweating. The loss of this substance may lead to attacks of cramps, unless the salts are replaced in the drinking water. In order to relieve both cramps and fatigue, Moss[4] recommends the addition of 6 grams of sodium chloride and 4 grams of potassium chloride in a gallon of water.

The deleterious physiologic effects of high temperatures exert a powerful influence upon physical activity, accidents, sickness and mortality. Both laboratory and field data (Tables 1 and 2) show clearly that physical work in warm atmospheres is a great effort, and that production falls progressively as the temperature rises. The incidence of industrial accidents reaches a minimum at about 68 F, increasing above and below that temperature (Fig. 1). Sickness and mortality rates increase progressively as the temperature rises. (Tables 3 and 4).

Effects of Cold

There is good evidence showing that exposure to cold and to changes in temperature lowers the resistance of animals to infection apparently by depressing the defensive mechanism of the organism. The prevalence of respiratory diseases in cold weather is attributed partly to the lowered resistance of the mucous membranes of the nose and throat which results from the vasomotor shifts of blood to the internal organs.[5] Keysser[6] believes that the lowered resistance is due to a diminution in the number and phagocytic activity of the leucocytes (white blood cells) brought about by exposure to cold and to changes in temperature.

On exposure to cold the loss of heat is increased considerably and only within certain limits is compensation possible by increased heat production and concentration of blood to the internal organs. The blood pressure rises owing to constriction in the peripheral vessels and to thickening of the blood. The skin subcutaneous tissues and muscles form reservoirs for storing the water which leaves the blood. In extremely cold atmospheres, the protective responses fail, and death may follow when the body temperature (rectal) falls down to about 80 F.

Cannon[7] showed that excessive loss of heat is associated with increased activity of the adrenal medulla. The extra output of adrenin hastens heat production which protects the organism against cooling. Bast[8] found a degeneration of thyroid and adrenal glands upon exposure to cold, and some physicians believe that the

[3] Influence of Effective Temperature Upon Bactericidal Action of Gastro-Intestinal Tract, by Arnold and Brody (Proceedings *Society Exp. Biol. Med.*, Vol. 24, 1927, p. 832).

[4] Some Effects of High Air Temperatures Upon the Miner, by K. N. Moss (Transactions *Inst. of Min. Eng.*, Vol. 66, 1924, p. 284).

[5] North, C. E.: Seasonal Diseases and Seasonal Temperatures. Amer. Jour. Public Health, 1913, 3, 222.

[6] Keysser, F.: Ztschr. fuer Balneol., Klimatol. U. S. W., 1913, 4, 421.

[7] Studies on the Condition of Activity of Endocrine Glands, by W. B. Cannon, A. Guerido, S. W. Britton and E. M. Bright (*American Journal Physiology*, Vol. 79, 1926, p. 466).

[8] Studies in Exhaustion Due to Lack of Sleep, by T. H. Bast, J. S. Supurnaw, B. Lieberman and J. Munro (*American Journal Physiology*, Vol. 85, 1928, p. 135).

common use of tonics in spring has something to do with the degeneration of the endocrine glands.

Effects of Changes in Temperature

In some unknown way, changes in temperature within the limit of adaptability of the human body seem to be stimulating and highly desirable. On the other hand, sudden changes from a warm to a cold atmosphere appear to be annoying and harmful to the mucous membranes of the upper respiratory tract. They produce a constriction of the blood vessels of the palate, throat, and tonsils, and diminish the phagocytic action of the white blood cells. This anaemic condition is highly favorable for the growth of pathogenic bacteria, and it may lead to *colds,* chronic atrophic rhinitis, etc.

Sickness records in industries seem to strengthen this belief. The Industrial Fatigue Research Board of England [9] found that in workers exposed to high temperatures and to changes in temperature, namely, steel melters, puddlers, and tin plate millmen, there is an excess of all sickness, the excess among the puddlers being due chiefly to respiratory diseases and rheumatism. The causative factor was not the heat itself but the sudden changes in temperature to which the workers were exposed. The tin plate millmen who were not exposed to chills, since they work almost continuously throughout the shift, had no excess of rheumatism and respiratory diseases. On the other hand, the blast furnacemen, who work mostly in the open, showed more respiratory sickness than the steel workers. This experience in British factories is well in accord with the findings in American industries.[10] According to these data the highest pneumonia death rate is associated with dust, extreme heat, exposure to cold, and to sudden changes in temperature.

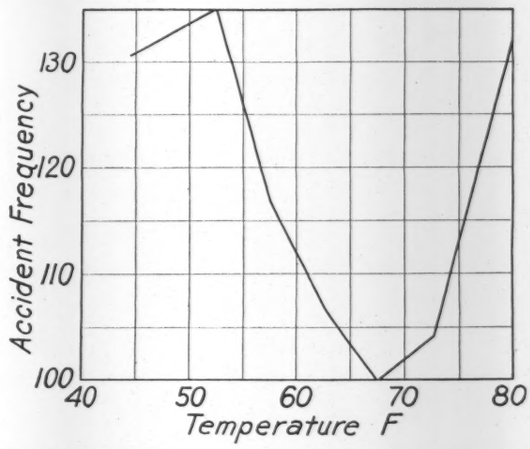


FIG. 1—FREQUENCIES OF ACCIDENTS IN MUNITION FACTORIES[a]

[a] Sixth Annual Report of the Industrial Fatigue Research Board, 1925, London.

TABLE 2—WORKING CAPACITY OF COAL MINERS IN RELATION TO EFFECTIVE TEMPERATURE[a]

EFFECTIVE TEMPERATURE	WORKING TIME (MIN PER HR)	TUB-FILLING TIME (MIN)	RATE OF PRODUCTION (PER CENT)
65.8	52.7	8.0	100
65.8	53.3	8.6	94
75.3	51.0	8.5	91
75.8	50.0	9.2	82
77.8	48.9	9.1	81
81.2	37.6	9.6	59

[a] The Relation of Atmospheric Conditions to the Working Capacity and the Accident Rate of Coal Miners, by H. M. Vernon (*Industrial Fatigue Research Board,* Report No. 39, 1927, London).

TABLE 3—SICKNESS AND ACCIDENT RATES OF COAL MINERS IN RELATION TO DRY-BULB TEMPERATURE[a]

TEMPERATURE RANGE	SHIFTS LOST ON ACCOUNT OF SICKNESS (PER CENT)	ACCIDENTS PER 100,000 MAN SHIFTS
Under 70	3.02	78
70–79	4.49	105
80 and Over	4.89	137

[a] A Study of Absenteeism in a Group of Ten Colleries, by H. M. Vernon, T. Bedford and C. G. Warner (*Industrial Fatigue Research Board,* Report No. 51, 1928, London).

TABLE 4—UNDERGROUND AIR TEMPERATURE (DRY BULB) IN RELATION TO DEATH RATES IN MINES OF THE WITWATERSRAND[a]

RANGE OF TEMPERATURE (DEG FAHR)	NUMBER OF MINES IN EACH TEMPERATURE GROUP	TOTAL DEATH RATE IN MINES OF EACH GROUP	PNEUMONIA DEATH RATE
66–68	2	13.1	5.1
69–71	14	15.7	6.9
72–74	20	16.4	7.6
75–77	9	17.9	6.5
78–80	8	20.5	8.5

[a] The Problem of Ventilation on the Witwatersrand, by E. H. Cluver (*Medical Journal,* 1922).

[9] Fatigue and Efficiency in the Iron and Steel Industry, by H. M. Vernon (Industrial Fatigue Research Board, Report No. 5, 1920, London).

Optimal Air Conditions for Comfort and Health

The Optimum Temperature

Comfort standards, in so far as temperature and humidity are concerned, are not absolute but are considerably affected by such factors as climate, adaptation, state of health, age, sex, clothing and physical activity. The variation is from 100 F for congenitally diseased premature infants (Table 5), to about 68 F for normal and healthy adults.

TABLE 5—AIR CONDITIONING REQUIREMENTS OF PREMATURE INFANTS AT INFANTS' HOSPITAL (BOSTON) 1926-1929[a]

WEIGHT GROUPS (LB)	NORMAL PREMATURE INFANTS — ROOM TEMPERATURES, F (DEW-POINT, 64 F; AIR MOVEMENT, 15 FPM)			CONGENITALLY DISEASED INFANTS — INCUBATOR TEMPERATURES, F (RELATIVE HUMIDITY OF ROOM, 65%; AIR MOVEMENT, 15 FPM)		
	MINIMUM	MAXIMUM	AVERAGE	MINIMUM	MAXIMUM	AVERAGE
1.5–1.9	83	88	86.0	86	100	93.5
2.0–2.4	82	87	84.5	86	98	90.5
2.5–2.9	78	86	83.0	84	95	88.0
3.0–3.4	77	86	80.5	79	90	83.5
3.5–3.9	75	85	78.5	79	88	81.0
4.0–4.4	75	85	77.0	78	86	79.0
4.5–4.9	75	85	76.0	76	85	77.0
5.0 and over	.....	80	.....	.....	85	.....

[a] Application of Air Conditioning to Premature Nurseries in Hospitals, by C. P. Yaglou, Philip Drinker and K. D. Blackfan, (*Heating, Piping and Air Conditioning,* July, 1930).

[10] Iron Foundry Workers Show Highest Percentage of Deaths from Pneumonia (Statistical Bulletin, Metropolitan Life Insurance Company, 1928, No. 3, 1).

Seasonal Variation of Comfort Zone

The comfort zone and its seasonal variation for adults between 20 and 70 years of age living in the northeastern parts of the United States is shown in Fig. 2.

It can be seen that the majority of the experimental subjects preferred an effective temperature of 66 F for comfort in winter time, and one of 71 F for summer comfort. The variation from winter to summer is probably due partly to adaptation to seasonal weather and partly to differences in the clothing worn in the two seasons. Young men as a general rule prefer conditions in the cool region of the comfort zone and women and older people in the warm region of the comfort zone.

Both summer and winter comfort zones shown in Fig. 2 make proper allowance for adaptation, and they apply to homes, offices, schools, and other similar places where people of sedentary habits spend from three to eight or more hours daily. In theaters and department stores, which are cooled artificially in warm weather, the contrast between outdoor and indoor air conditions becomes the deciding factor in regard to the temperature and humidity to be maintained. If the contrast is too great, unpleasant sensations of chill or of intense heat and mugginess will be experienced in entering and leaving the building. This unpleasant situation may be avoided by increasing the indoor temperature to a level between the optimum dry bulb for continuous exposure (76 F) and that prevailing out of doors, and by decreasing the relative humidity to 60 per cent or less, in order to keep the skin and clothing dry. For economical reasons the relative humidity in air-conditioned buildings is, as a general rule, too high in summer and too low in winter, whereas for comfort it should be just the opposite. In all cases, the change in humidity should be accompanied by an inverse change in temperature in order to allow for the prevailing outdoor conditions. Desirable indoor air conditions, for sedentary individuals, corresponding to various outdoor temperatures are given in Table 6.

TABLE 6—DESIRABLE INDOOR TEMPERATURES IN SUMMER CORRESPONDING TO OUTDOOR TEMPERATURES

DEGREES OUTSIDE	DEGREES INSIDE		
DRY-BULB	DRY-BULB	WET-BULB	EFFECTIVE TEMPERATURE
95	80.0	65.2	73.4
90	78.0	64.5	72.2
85	76.5	64.0	71.1
80	75.0	63.5	70.2
75	73.5	63.0	69.3
70	72.0	62.5	68.2

Comfort Zones for Working Men. Data on comfort zones for men at work are meager. Satisfactory air conditions are found to vary from 40 deg to 70 deg E T, depending upon the rate of work and amount of clothing worn. In hot industries, an effective temperature of 80 deg may be set as the upper limit compatible with efficiency and, whenever possible, this should be reduced to 70 deg or less.

Since the prolonged effects of temperature, humidity and air movement on health are not so well known as their effects on comfort, the optimum conditions for health may not be identical with those for comfort. On general physiologic grounds, however, it seems reasonable to assume for the present that the two do not differ greatly, since this is in accordance with the efficient operation of the heat regulating mechanism of the body. This belief is strengthened by results of studies on premature infants over a four-year period[11]. By adjusting the temperature and humidity so as to stabilize the body temperature of these infants, the incidence of diarrhea and mortality was automatically decreased, gains in body weight increased and infections were reduced to a minimum.

The Optimum Humidity

The effect of humidity on comfort and health has not been satisfactorily established. There is no consistent research to show the response of adult human beings to controlled humidity conditions over a prolonged period of time. It is becoming more and more apparent however, that humidity exerts a considerable influence on the organism. A low temperature with a comparatively high humidity is as a general rule more comfortable and probably more healthful than a high temperature with very low humidity. Extremes in humidity are undesirable because they produce a disturbance of water balance and evident symptoms of restlessness. Too dry air desiccates the skin and mucous membranes of the upper respiratory tract and this is believed to play a role in the incidence of respiratory disorders.[12, 13, 14] On the other hand, when the relative humidity exceeds 85 per cent, the organism cannot give off readily its latent heat. This latter effect is manifested not by a rise in body temperature, but by a decrease of about 20 per cent in the general metabolism.[15]

For the premature infant, a high relative humidity of about 65 per cent is demonstrably beneficial to health and growth, and according to Huntington[16], this seems to be the case for adults also. All of these studies indicate that the optimum humidity must always be considered in combination with temperature.

The chief drawback to high indoor humidities is condensation and frosting on the window glass. Unless double glazing, or some other method, is used to prevent condensation, it is desirable to assume in general air conditioning work a relative humidity range of 40 per cent to 65 per cent, the former representing a maximum winter condition and the latter a maximum summer condition. Experiments show that with an outdoor temperature of 20 F, condensation will occur on a single glass when the relative humidity reaches 30 per cent, and on double glass when the humidity reaches 60 per cent. In zero weather double glazing will allow an

[11] Application of Air Conditioning to Premature Nurseries in Hospitals, by C. P. Yaglou, Philip Drinker and Kenneth Blackfan (*Heating, Piping and Air Conditioning*, July, 1930).

[12] Reactions of the Nasal Cavity and Post-Nasal Space to Chilling of the Body Surface, by Mudd, Stuart, et al. (*Journal Experimental Medicine*, 1921, 34, 11).

[13] Reactions of the Nasal Cavity and Post-Nasal Space to Chilling of the Body Surfaces, by A. Goldman, et al. II. Concurrent Study of Bacteriology of Nose and Throat (*Journal Infectious Diseases*, 1921, 29, 151).

[14] The Etiology of Acute Inflammations of the Nose, Pharynx and Tonsils, by Mudd, Stuart, et al. (*Am. Otol., Rinol., and Laryngol*, 1921).

[15] Lefevre, J., and Auguet, A.: Effect of Humidity of the Air on Metabolism. *Compt. Rend.*, 1930, 190, 326-8.

[16] Weather and Health, by E. Huntington (Bulletin of the National Research Council No. 75. *The National Academy of Science*, Washington, D. C., 1930).

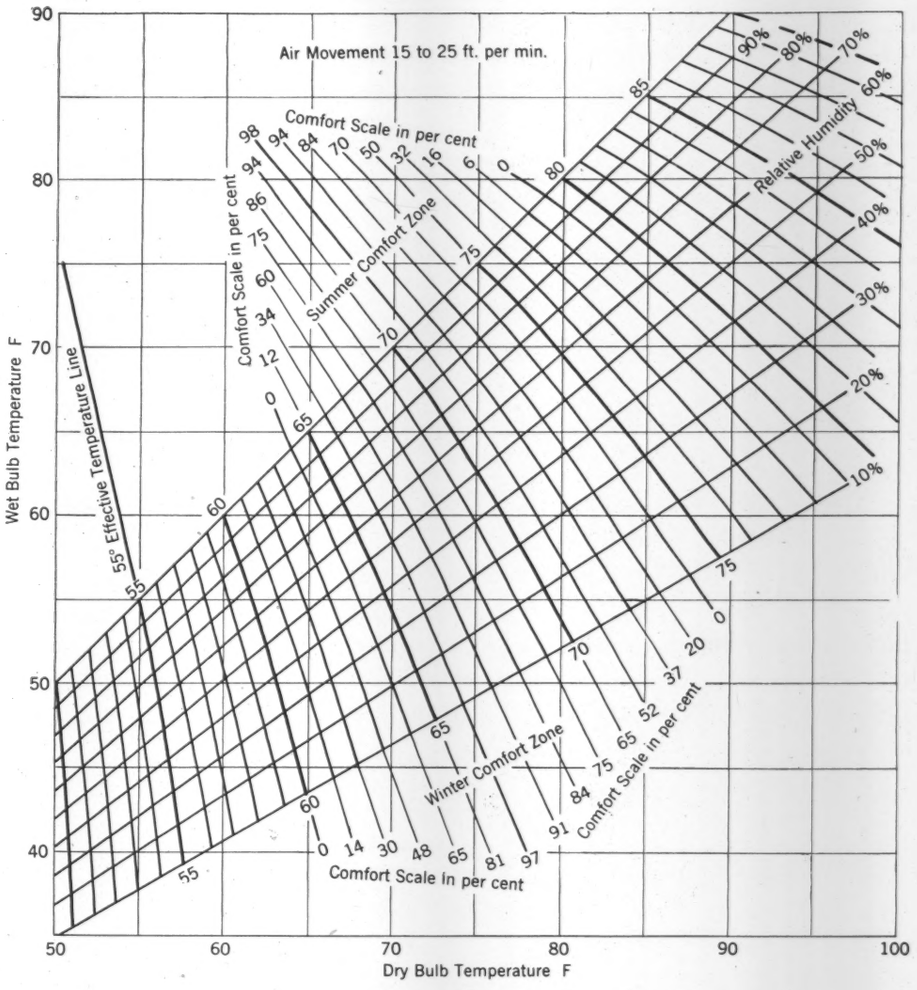


FIG. 2—SEASONAL VARIATION IN THE COMFORT ZONE [a]

[a] See Determination of the Comfort Zone, by F. C. Houghten and C. P. Yaglou. (A. S. H. V. E. TRANSACTIONS, Vol. 29, 1923). Also, The Summer Comfort Zone; Climate and Clothing, by C. P. Yaglou and Philip Drinker (A. S. H. V. E. TRANSACTIONS, Vol. 35, 1929).

indoor humidity of 50 per cent before condensation begins.

Recirculation and the Use of Ozone

Recirculation of air to the extent of 50 per cent or more effects a considerable saving in both heating and refrigeration, but as a general rule it impairs the quality of the air by excessive humidity, excessive odors or both. Moreover, recirculation has a tendency to deprive the air of its ionic content[17], but the influence of this factor on comfort and health remains to be proven.

The popular belief regarding the alleged virtues of ozone in recirculated air is debatable and not well founded. Numerous researches[18, 19, 20] have shown that ozone in concentrations permissible in air conditioning work (0.1 to 0.5 parts per million parts of air) exerts no effect whatever on pathogenic air-borne organisms and it does not destroy odors, but it merely masks them by olfactory compensation. It requires at least 13 parts of ozone per million parts of air to influence bacteria[21]. Since human beings are injuriously affected by ozone in concentrations of one part or more per million parts of air, and since this amount will not destroy odors, nor kill bacteria, nor purify organic matter, the value of ozone in air conditioning is very doubtful.

Ultra-Violet Radiation and Ionization in Relation to Air Conditioning

In spite of rapid advances in air conditioning during the past few years, the secrets of reproducing indoors the natural climatic elements as they exist in open country under ideal weather conditions have not as yet been fully ascertained. Extensive studies have failed to discover the stimulating quality in open country air which is apparently lost when the air is brought indoors, and particularly when it is treated mechanically. Ultra violet light and ionization have been suggested but have not yet been identified.

It is generally recognized that total solar radiation in open air is a powerful germicidal and an excellent curative agent. A critical review of the literature, however, discloses that artificial ultra violet radiation has little importance in air conditioning, at least for the time being, because it is not known which of the solar rays brings about the cure, and what physiologic processes are involved in their action. With the exception of rickets and certain skin diseases, artificial radiation was found to have no effect on susceptibility, incidence, and

[17] Changes in Ionic Content in Occupied Rooms, Ventilated by Natural and Mechanical Methods, by C. P. Yaglou, L. C. Benjamin and S. P. Choate (*Heating, Piping and Air Conditioning*, October, 1931).

[18] Proc. Royal Society of London, by L. Hill and M. Flack (1911, B, Vol. 84, p. 404).

[19] Jordan and Carlson (Journal *American Medical Association*, 1913, Vol. 61, p. 1607).

[20] Konrich (*Ztschr.* f. Hyg., 1913, Vol. 73, p. 443).

[21] Preventive Medicine and Hygiene, by Milton J. Rosenau (D. Appleton and Company, New York, 1927).

resistance to respiratory infections, no permanent effects on blood, and no effect on the general metabolism of human beings[22].

Ionization, on the other hand, seems to offer a fruitful field for research. Recent experiments[23] show that in occupied rooms there is a marked decrease in both positive and negative small ions. As shown in Fig. 3, the ionic content fell abruptly soon after the occupants assembled to a very low level which was maintained until the occupants left the rooms. Both positive and negative ions began to rise again as soon as the occupants departed. The problem now is to determine whether such alterations in the electrical quality of air have any significant bearing on comfort and health. If this proves to be the case, some artificial source of ionization must be employed in occupied rooms.

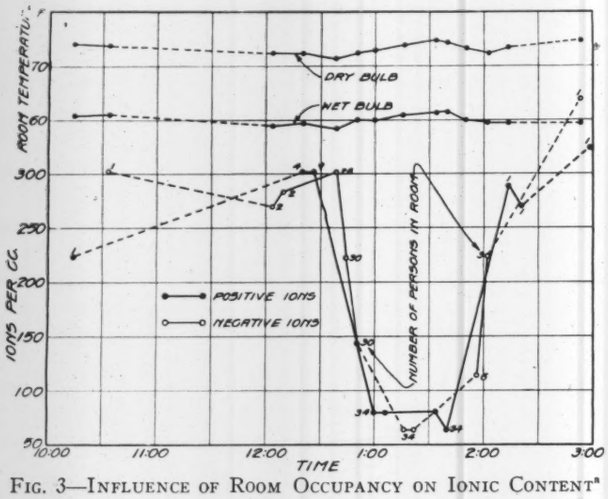


FIG. 3—INFLUENCE OF ROOM OCCUPANCY ON IONIC CONTENT[a]

[a] Room Dimensions 30 ft x 33 ft x 10 ft 6 in. Number of Occupants, Thirty-four. Good Window-Gravity Ventilation.

[b] See Changes in Ionic Content of Air in Occupied Rooms Ventilated by Natural and by Mechanical Methods, by C. P. Yaglou, L. Claribel Benjamin, and Sarah P. Choate, Journal of the Society (*Heating, Piping and Air Conditioning*), October, 1931.

Acclimatization and the Psychologic Factor in Relation to Air Conditioning

Acclimatization and the factor of psychology are two important influences in air conditioning which cannot be ignored. The first is man's ability to adapt himself to changes in air conditions; the second is an intangible matter of habit and suggestion.

Some persons regard the unnecessary endurance of cold as a virtue. They believe that the human organism can adapt itself to a wide range of air conditions with no apparent discomfort or injury to health. In the light of our present knowledge of air conditioning these views are not justified. Acclimatization to extreme conditions involves a strain upon the heat regulating system and it interferes with the normal physiologic functions of the human body. Thousands of years in the heat of Africa do not seem to have acclimatized the Negro to a temperature averaging 80 F. The same holds true of northern races with respect to cold, although the effects are mitigated by artificial control. All this seems to indicate that adaptation to a climate averaging between 60 and 75 F is a very primitive trait.[24]

In certain individuals the psychologic factor is much more powerful than acclimatization. A fresh air fiend may suffer in a room with windows closed no matter how good the ventilation system. As a matter of fact, instances are known in which paid subjects refused to stay in a windowless but properly conditioned experimental chamber because the atmosphere felt suffocating to them upon entering the room.

The Controversy Between Natural and Mechanical Ventilation

The extensive studies of the N. Y. S. C. V. have brought out clearly the difficulties involved in obtaining reliable evidence on the relative merits of natural and mechanical ventilation from the health standpoint. The first studies of the Commission [25, 26, 27, 28], showed quite forcibly that in schoolrooms ventilated by mechanical methods, the incidence of respiratory illness was higher than in rooms ventilated by window-gravity methods. The excess was ascribed to a slight overheating in the mechanically ventilated rooms on account of the greater air supply for which these systems are designed.

More recent studies of the Commission, however [29, 30, 31], have proved beyond doubt that the apparent superiority of window-gravity methods was due partly to the inadequate criterion used for the healthfulness of air conditions, and partly to uncompensated variables, such as nationality, sex, age, social and economic status of pupils, and distance travelled by the pupils to reach school. When these variables were compensated by the use of comparable groups of pupils, and when records of respiratory illness among comparable groups were made by the same observer, the type of ventilation, whether natural or mechanical, ceased to be a factor, in so far as incidence of respiratory disease was concerned.

The final conclusions of the Commission [32] are: (*a*) that neither the type of ventilation nor the temperature (within the ordinary range), nor the rate of air change affect respiratory illness among pupils, and (*b*) that equally good air conditions may be secured by either plenum ventilation, unit ventilation, or window-gravity methods, with an air supply of either 30 or 15 cfm per pupil.

Substantially similar conclusions have been reached by

[22] Light—Its Photodynamic Activity and Use as a Therapeutic Agent, by F. W. Schultz, K. W. Stenstrom and E. M. Clausen (Report to the White House Conference on Child Health and Protection, Washington, February, 1931).

[23] Changes in Ionic Content in Occupied Rooms, Ventilated by Natural and Mechanical Methods, by C. P. Yaglou, L. C. Benjamin and S. P. Choate (*Heating, Piping and Air Conditioning*, October, 1931).

[24] Civilization and Climate, by Ellsworth Huntington (Yale University Press, New Haven, 1924).

[25] Ventilation—Report of the New York State Commission on Ventilation (E. P. Dutton and Company, New York, 1923).

[26] Effects of Mechanical and Natural Ventilation on the Health of School Children, by Thomas J. Duffield (A. S. H. V. E. TRANSACTIONS, Vol. 34, 1928, 109).

[27] A Study of the Relationship Between Type of Ventilation and Respiratory Illness in Certain Schools of New Haven, Connecticut, by L. Greenburgh (Public Health Reports, 1929, 64, 285).

[28] Symposium on Schoolroom Ventilation, by L. W. Childs (Reports on Schoolroom Ventilation Studies, *American Journal Public Health*, 1929, 19, 59).

[29] A Study of Ventilation and Respiratory Illness in Syracuse Schools, with an Analysis of Factors Affecting Criteria Used, by Rufus Cole, et al. (*The American Journal Hygiene*, Vol. 12, 1930, 196).

[30] A Study of Ventilation and Respiratory Illness in Syracuse Schools; Rate of Air Flow and Room Temperature in Relation to the Health of School Children, by Rufus Cole, et al. (*American Journal of Hygiene*, Vol. 12, 1930, 215).

[31] A Study of Ventilation and Respiratory Illness in New York Schools; Comparison of Window-Gravity Ventilation and of Unit Fan Ventilation with Varying Air Flow, by Rufus Cole, et al. (*Ibid.*, Vol. 13, 1931, 235).

[32] Winslow, C.-E. A., et al. School Ventilation Principles and Practices. (Final Contribution.) Bureau of Publications, Teachers College, Columbia Univ., N. Y. C., 1931.

Vernon [33] in reviewing older studies in England. The findings of the various observers were so contradictory that they virtually cancelled one another.

According to the present knowledge of the requirements of ventilation, it is fair to say that window-gravity ventilation will prove satisfactory for many school rooms of the type studied by the Commission. Mechanical ventilation must be resorted to in large classrooms, offices, auditoriums, and factories, or where noise, dust, odors, etc., make it inadvisable to keep the windows open.

Air Pollution in Relation to Comfort and Health

General Atmospheric Pollution

Temperature, humidity and air movement seem, as a general rule, of major importance to comfort and health, but under certain circumstances in which dusts and toxic gases are conspicuously present in the atmosphere, the factor of air purity becomes of utmost importance. Many kinds of dusts and gases are capable of producing profound pathologic effects and damage to property and vegetation. The harmful effects which may result depend largely upon the chemical and physical nature of the impurities, the concentration, length of time, and conditions under which they are breathed.

It is intimated by physicians and others that general air pollution in many industrial cities is provocative of pulmonary diseases and that under certain circumstances it may even lead to cancer through biochemical irritation of some unknown underlying cause. The constant inhalation of solid and gaseous ingredients in city air (carbon, silica, sulphur dioxide, etc.) irritates the mucous membrane of the nose, throat and lungs producing discomfort and in some instances, causes injury to health. Smoke is not directly poisonous and it probably has no deleterious influence on tuberculosis, but it appears to have some influence on acute lung diseases. It may actually give rise to *anthracosis* (a dark staining of the tissue cells lining the alveoli of the lungs) and thus impair the function of the lung cells, predisposing to pneumonia.

In addition to these direct effects, dusty air has important indirect effects upon health and property. The loss of light, particularly the occlusion of solar ultra-violet light due to smoke and soot, is beginning to be recognized as a health problem in many industrial cities. Measurements of solar radiation in Baltimore by actinic methods show that the ultra-violet light in the country was 50 per cent greater than in the city. [34] In New York City Ives found a loss as great as 50 per cent in visible light using the photoelectric cell method.[35] Although these facts are generally recognized, there are no data to prove that the effect of dust and smoke in ordinary air is sufficient to undermine demonstrably the health of city dwellers. The objection is considered to be aesthetic and largely economic. On the other hand, *hay fever* and other allergic diseases caused by air-borne pollen, are excellent illustrations of the possible harmful effects of solid suspensoids of a specific nature.

[33] Vernon, H. M.: Ventilation in Relation to Health. Jour. of State Med., 1926, 34, 12, 683.

[34] Shrader, J. H., Coblentz, M. H., and Korff, F. A.: Effects of Atmospheric Pollution Upon Incidence of Solar Ultra-Violet Light. Amer. Jour. of Public Health, 1929, 19, 7.

[35] Ives, J. E.: Studies in Illumination. U. S. Public Health Service, 1930, Bulletin No. 197.

Industrial Air Pollution

In many industrial processes, sufficient amounts of dusts, fumes and vapors are liberated to be injurious to the health of the workers. Some dusts are poisonous (lead, mercury, arsenic, manganese, cadmium, etc.) and some act as irritants (silica, steel, iron, granite, etc.). Certain dusts may produce catarrhal conditions and increase susceptibility to such diseases as bronchitis, pneumonia and tuberculosis. Silicious dust is especially harmful because it has a direct damaging action upon the tissue of the lungs. On the other hand, organic dusts, both animal and vegetable (hair, pollen, textile fibers, etc.), do not seem to affect the lungs at all, although they may cause considerable discomfort to persons sensitive to them by their action on the upper respiratory passages.

Industrial gases and fumes act specifically upon the mucous membranes, the lungs, blood, skin, and eyes. Some of these substances are extremely poisonous and their action sets in after very short exposures, as is true, for example, in the inhalation of carbon monoxide, hydrogen sulphide, ammonia, chlorine, bromine, arsene and cyanogen.

The industrial processes which liberate harmful substances are too manifold and the effects too diverse to be considered here. It would suffice to discuss the commonest and most serious with which the ventilating engineer may be confronted, namely, carbon monoxide, lead, and silica. For a more thorough treatise on the subject the reader is referred to books by Hamilton,[36] Rosenau,[37] and Henderson and Haggard.[38]

Carbon Monoxide is one of the commonest forms of gas poisoning. It is met with in mines, foundries, coke oven sheds, garages, and in houses. The poisonous action is due to the fact that the combining power of carbon monoxide with the hemoglobin of the red blood corpuscles is about 300 times greater than that of oxygen. This stable combination destroys the power of the hemoglobin to unite with oxygen in the lungs and to supply it to the tissues. The effects are, therefore, due to lack of oxygen and the symptoms are those of anoxemia, namely, dizziness, headache, sleepiness, fatigue, and in extreme cases, paralysis and death. The dangerous saturation level of the blood with carbon monoxide is about 50 per cent. Even as little as 0.07 per cent in air will render, in half an hour, one quarter of the red corpuscles incapable of uniting with oxygen. One to two parts per 10,000 parts of air is set as a safe limit of pollution which may be breathed for a long time without producing perceptible symptoms.

Silicosis is a chronic disease of the lungs which results from local physio-chemical action of hydrated silica upon the pulmonary tissue causing progressive lymphatic fibrosis and rendering the tissue susceptible to tuberculosis. The disease is slow in evolution, requiring usually a number of years of exposure. It occurs principally among granite workers, sand blasters, metal miners, metal polishers, potters, millstone workers and others.

Studies have shown that dust containing a high per-

[36] Industrial Poisons in the United States, by Alice Hamilton (The MacMillan Company, New York, 1925).

[37] Preventive Medicine and Hygiene, by Milton J. Rosenau (D. Appleton and Company, New York, 1927).

[38] Noxious Gases, by Y. Henderson and H. Haggard (The Chemical Catalog Company, New York, 1927).

centage of free silica in particles less than 10 microns (1 micron = 0.0004 in.) in size are chiefly responsible for silicosis because only these particles reach the depths of the lungs. Workers exposed to air containing about 15,000,000, or less, dust particles per cubic foot (35 per cent free silica in the form of quartz) under 10 microns in size, showed practically no sickness from tuberculosis. In order to attain this safe concentration a velocity of 1500 fpm was necessary in the suction hoods [39] intended for dust removal.

Lead poisoning is the most insidious and most frequent of all of the occupational intoxicants. It occurs principally among lead workers and smelters, lead miners, potters, painters, typesetters, stereotypers, plumbers, glass, gold and silver workers. Lead, in practically all forms, is a cumulative poison which is absorbed by way of the blood stream, chiefly from the respiratory tract, but also from the digestive tract and from the skin. The effect may be acute or chronic poisoning. The principal symptoms are colic, constipation, anemia, headache, anorexia, a blueish line along the edges of the gums, rheumatic pains, and in extreme conditions, paralysis, blindness, insanity and death.

It has been found [40] that 2 milligrams per day is the lowest dose, by inhalation, which in the course of years may result in lead poisoning. Regular inhalation during the usual working hours of air containing less than 0.2 mg of lead per cubic meter does not seem to produce serious lead intoxication in individuals of representative industrial groups.[41]

The prevention of industrial hazards from dusts and poisonous gases is largely a ventilation problem, and it consists in keeping the impurities in air down to a safe concentration. At present, there are no accurate standards on which to base the design of the ventilation equipment. Approximate data on the toxicity of various gases and fumes met with in industrial establishments are given in Table 7. The maximum allowable concentrations for prolonged exposure (Column 5) will, perhaps, in the course of weeks, months or years, produce pernicious effects because the experiments cited here were limited in most cases to exposures of not more than a week.

Very little is known concerning the physiologic effects induced by various concentrations of dusts and by various durations of exposure. With very few exceptions, ventilation standards for dusts are lacking and the only way to judge the safe concentration is by means of periodic medical examinations of the workers and by routine measurements of dust concentration.

Table 7—Toxicity of Gases and Fumes in Parts per 10,000 Parts of Air [a]

Vapor or Gas	Rapidly Fatal	Maximum Concentration for from ½–1 hour	Maximum Concentration for 1 hour	Maximum Allowable for Prolonged Exposure
Carbon monoxide	40	15–20	10	1
Carbon dioxide	800–1000	.....	.....	.....
Hydrocyanic acid	30	1½	½	1/5
Ammonia	50–100	25	3	1
Hydrochloric acid gas	40–20	½	.....	1/10
Chlorine	10	½	.....	1/100
Hydrofluoric acid gas	2	1/10	.....	1/33
Sulphur dioxide	4–5	½–1	.....	1/10
Carbon bisulphide	.....	11	5	½
Hydrogen sulphide	10–30	5–7	2–3	1
Phosphene	20	4–6	1–2	.....
Arsine	2½	½	½	.....
Phosgene	Over ¼	¼	.....	1/100
Nitrous fumes	2½–7½	1–1½	.....	1/3
Benzene	190	.....	31–47	1½–3
Toluene and xylene	190	.....	31–47	.....
Aniline	.....	.....	1–1½	1/10
Nitrobenzene	.....	.....	1/100	1/500
Petrol	243	100–220	.....	.....
Carbon tetrachloride	480	240	40	16
Chloroform	250	140	50	2
Tetrachlorethane	73	.....	.....	1½
Trichlorethylene	370	.....	.....	.....
Methyl chloride	1500–3000	200–400	70	5–10
Methyl bromide	200–400	20–40	10	2
Lead vapor	.....	.....	.....	5–6

[a] Original data compiled by Y. Henderson and H. Haggard (See Noxious Gases published by the Chemical Catalog Company, New York, 1927). Data revised by T. M. Legge. (See Lessons Learned from Industrial Gases and Fumes, published by the *Institute of Chemistry of Great Britain and Ireland*, London, 1930).

[39] A Study of the Efficiency of Dust Removal Systems in Granite-Cutting Plants, by J. J. Bloomfield (Public Health Reports No. 44, 1929, 42, 2505).

[40] Lead Poisoning, by Thomas Morrison Legge (Journal *Royal Society Arts.*, Vol. 77, 1929, p. 1023).

[41] What Is a Dangerous Quantity of Lead Dust in Air, by C. M. Salls (Industrial Hygiene Bulletin, New York State Department of Labor, 1925, 2 and 3, 9).

British Air-Conditioning Hut Makes Climate to Suit Needs of Workers

In the Borough of Westminster, London, is the building which houses the Home Office Industrial Museum. Though only three years old, its value is gradually becoming more widely known, and last year no fewer than 150 organized parties visited the museum, some of them numbering more than 300 persons. Many of these parties went for the express purpose of experiencing a demonstration of the air-conditioning hut.

The object of the hut is to enable visitors to experience the immediate effects of different atmospheric conditions and to show how these are influenced by the variation of certain factors. When the demonstration begins, the air in the hut is cool and still. By means of a fan the air is made to circulate around the passages in which the visitors stand. The effect of the air movement is to produce a sensation of coolness, although the actual temperature is unchanged. As the velocity of the air movement is increased, the coolness is increased until eventually discomfort is produced.

Then electrically-heated panels are put into operation which compensate for the body heat given up to the moving air. This combination of radiant heat with air at a comparatively low temperature moving at a reasonable velocity produces a comfortable condition. When steam jets are turned on, the humidity of the air is increased. This is done with fan stopped, and discomfort is felt until the fan puts matters right again.

Thus the advantages and disadvantages of the various conditions of the air can be shown, and employers of labor in factories can see how the best conditions for work can be effected.

NOMINATIONS FOR 1932

The Nominating Committee appointed to select candidates for Officers of the Society for the coming year, 1932, takes pleasure in submitting the following list of nominees:

For President:

F. B. ROWLEY, Minneapolis, Minn.

For First Vice-President:

W. T. JONES, Boston, Mass.

For Second Vice-President:

C. V. HAYNES, New York, N. Y.

For Treasurer:

F. D. MENSING, Philadelphia, Pa.

For Members of the Council:

Three-Year Term

F. E. GIESECKE, College Station, Texas.
G. L. LARSON, Madison, Wis.
J. F. MCINTIRE, Detroit, Mich.
W. E. STARK, Cleveland, Ohio

Respectfully submitted,

NOMINATING COMMITTEE

R. C. BOLSINGER, *Chairman*

Chapters	*Representative*
Cleveland	H. M. NOBIS
Illinois	H. M. HART
Kansas City	N. W. DOWNES
Massachusetts	DAVID MOULTON
Michigan	J. F. MCINTIRE
Minnesota	M. S. WUNDERLICH
New York	RUSSELL DONNELLY
Western New York	F. H. BURKE
Ontario	H. J. CHURCH
Pacific Northwest	E. O. EASTWOOD
Philadelphia	R. C. BOLSINGER
Pittsburgh	F. C. MCINTOSH
St. Louis	C. A. PICKETT
Southern California	R. L. GIFFORD
Wisconsin	E. A. JONES

In accordance with the provisions of the Amended Constitution, as given, ballots containing the names of the above candidates will soon be sent to the membership for voting upon prior to the Annual Meeting in January.

Art. 9—*Section* 2. The Nominating Committee shall consist of a member designated by each Chapter, which shall effect its own organization and elect its own Chairman.

Section 3. The Committee may meet at the call of the Chairman, for discussion, but the official meeting shall be held at the Semi-Annual Meeting of the Society, at which time the candidates for each of the offices of the Society to be filled at the next Annual Meeting shall be selected by a majority vote of the members of the Committee present and voting. Five shall constitute a quorum.

Section 4. The Committee shall first secure the consent of all candidates selected, and shall, if possible, announce the names of the candidates at the Semi-Annual Meeting, but in any case the names of the candidates shall be certified to the Secretary of the Society at least four months before the next Annual Meeting. The Secretary shall publish these names in the October issue of THE JOURNAL.

Art. 10.—*Section* 3. The Secretary shall prepare ballots with the names of all candidates and forward them to the members at least thirty (30) days before the date of the Annual Meeting.

Nominations for Members of Committee on Research

Although committees of the Society are usually appointed, in view of the great importance of the Committee on Research and the financial responsibility it would be called upon to assume, it has been made more representative of the entire membership of the Society by a process of *election.* The election is governed by the By-Laws for the election of officers, with the single exception, that Members of the Committee on Research are nominated by the Council instead of by a Nominating Committee.

In accordance with the Regulations for the government of the Research Laboratory, adopted at the January, 1919, Meeting, and revised January, 1930, the Council announces the nomination of the following members of the Committee for election to succeed those members whose present terms expire January, 1932:

D. E. FRENCH, Philadelphia, Pa.
F. E. GIESECKE, College Station, Texas
A. P. KRATZ, Urbana, Ill.
G. L. LARSON, Madison, Wis.
L. A. HARDING, Buffalo, N. Y.

The regulations governing the nomination and election of members of the Committee on Research are as follows:

SECTION I—ORGANIZATION

Research Committee

1. There shall be a standing committee known as the Research Committee, consisting of fifteen members, each serving for three years, and five retiring each year.

a. The Council shall nominate previous to October first of each year five members to fill the vacancies of those retiring at the next Annual Meeting.

b. The nominations made by the Council shall be published in the October issue of the Society's JOURNAL.

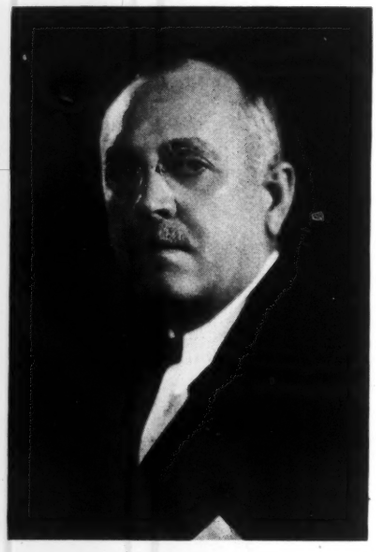
c. Prior to December first of any year, any ten members, over their own signatures, may nominate one or more additional members of the Research Committee, and such additional nominations shall be placed on the ballot opposite the nominations made by the Council.

d. The election shall otherwise conform to the regulations provided for the election of officers of the Society.

e. Vacancies may be filled by the Council, such persons chosen by the Council to serve until a successor is elected at the next Annual Meeting.

WALTER KLIE
Finance

C. F. EVELETH
Reception and Registration



T. A. WEAGER
General Chairman of Arrangements

R. G. DAVIS
Entertainment

W. C. KAMMERER
Banquet

F. H. MORRIS
Transportation

D. E. HUMPHREY

A. S. H. V. E.
Cleveland — January 25-29
A. S. R. E.

THE Cleveland Chapter of the A. S. H. V. E. will act as host for the 38th Annual Meeting of the Society to be held at the Hotel Statler, January 25-29, 1932, simultaneously with the 27th Annual Meeting of the *A. S. R. E.* in the Hotel Cleveland and the Second International Heating and Ventilating Exposition at the Cleveland Auditorium. These three events will attract thousands of members and guests to the technical sessions and exhibits.

Registration of Society members will commence on Monday, January 25, at the Hotel Statler and during the day the Council of the Society will meet, as well as such important committees as the Committee on Research and the Committee on Ventilation Standards. The Council will assemble during the morning to consider matters of unfinished business and in the evening will join the Council of the *A. S. R. E.* for dinner.

Technical Sessions

An extensive four-day program has been prepared with joint sessions of the A. S. H. V. E. and the *A. S. R. E.* on Wednesday and Thursday mornings.

Promptly at 9:30 a. m. on Tuesday, the 26th, President Carrier will call the 38th Annual Meeting to order and will hear reports from the various officers and committees. Two technical papers will be presented, both dealing with heat transmission

"AKRON" IN FLIGHT OVER GOODYEAR-ZEPPELIN FACTORIES

HOTEL STATLER
A. S. H. V. E. HEADQUARTERS

PROGRAM

38th ANNUAL MEETING

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

January 25-29, 1932
Hotel Statler, Cleveland, Ohio

Monday—January 25th

10:00 A. M.—Meeting of the Council

12:30 P. M.—Luncheon for Officers and Authors

2:00 P. M.—Meeting Committee on Research

7:00 P. M.—Joint Dinner-Meeting of the Councils of the A. S. H. V. E. and *A. S. R. E.* and their wives (Hotel Cleveland)

8:30 P. M.—Inspection of Second Heating and Ventilating Exposition

Tuesday—January 26th

9:30 A. M.—First Session (Ballroom Hotel Statler)

Address of Welcome
Response by Pres. W. H. Carrier
Reports of Officers
Report of Council
Reports of Council Committees
Executive, F. B. Rowley, Chairman
Finance, F. C. McIntosh, Chairman
Membership, Roswell Farnham, Chairman
Publication, W. A. Rowe, Chairman
Report of Committee on Research, C. V. Haynes, Chairman
Report of Director of Research Laboratory, F. C. Houghten
Reports of Technical Advisory Committees
Reports of Special Committees
Guide Publication Committee, D. S. Boyden, Chairman
Committee on Increase of Membership, C. W. Farrar, Chairman
Reports of Continuing and Cooperating Committees
Report of Tellers of Election

Technical Papers:

Surface Coefficients as Affected by Direction of Wind, by F. B. Rowley and W. A. Eckley
Conductivity of Concrete, by F. C. Houghten and Carl Gutberlet

2:00 P. M.—Second Session (Ballroom, Hotel Statler)

Technical Papers:

Importance of Radiation in Heat Transfer Through Air Spaces, by E. R. Queer
Heat Emission from Iron and Copper Pipe, by F. C. Houghten and Carl Gutberlet
Hot Water Friction Heads in One-Inch Cast Iron Tees, by F. E. Giesecke and W. H. Badgett
Some Fundamental Considerations of Corrosion in Steam and Condensate Lines, by R. E. Hall and A. R. Mumford

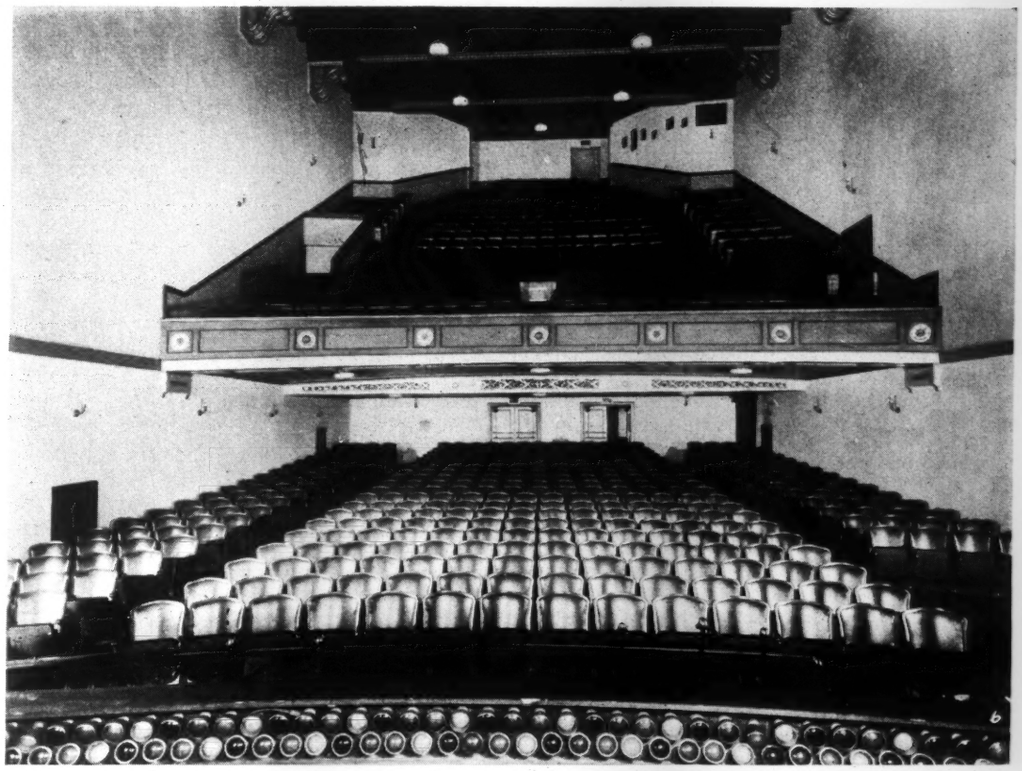
7:00 P. M.—Inspection of Second Heating and Ventilating Exposition

8:30 P. M.—Informal Reception including entertainment and Monte Carlo (Ballroom Hotel Statler)

Wednesday—January 27th

10:00 A. M.—Third Session (Little Theatre, Cleveland Auditorium)

Joint Meeting with *A. S. R. E.*—Program arranged by A. S. H. V. E.—Pres. W. H. Carrier presiding



LITTLE THEATER IN CLEVELAND PUBLIC HALL

Technical Papers:

Changes in Ionic Content of Air in Occupied Rooms Ventilated by Natural and by Mechanical Methods, by C. P. Yaglou, L. C. Benjamin and S. P. Choate

Acoustical Problems in Heating and Ventilating of Buildings, by V. O. Knudsen

Heat Transmission as Influenced by Heat Capacity and Solar Radiation, by F. C. Houghten, J. L. Blackshaw and E. M. Pugh

Field Studies of Office Building Cooling, by S. S. Sanford, E. P. Wells and J. H. Walker

2:00 P. M.—Trip to Akron (Goodyear-Zeppelin Hangar)

7:00 P. M.—Dinner for Past-Presidents and Wives (Hotel Statler)

8:00 P. M.—Ladies' Theatre Party at Keith's Palace

8:00 P. M.—Inspection of Second Heating and Ventilating Exposition

Thursday—January 28th

10:00 A. M.—Fourth Session (Little Theatre, Cleveland Auditorium)

Joint Session with *A. S. R. E.*—Program arranged by *A. S. R. E.*—Alvin H. Baer presiding

Technical Papers:

Application of Electric Refrigeration to the Heating and Cooling of Houses, by F. H. Faust, E. W. Roessler and A. R. Stevenson, Jr.

Air Conditioning Applied to Cold Storage Including a New Psychrometric Chart, by C. A. Bulkeley

Biological Aspects of Thermal Engineering, by Samuel R. Prescott

2:00 P. M.—Fifth Session (Ballroom, Hotel Statler)

Report of the Committee on Testing and Rating Unit Ventilators, John Howatt, Chairman

Report of Committee on Ventilation Standards, W. H. Driscoll, Chairman

2:30 P. M.—Ladies' Bridge (Chamber of Commerce)

7:00 P. M.—Annual Banquet and Dance (Hotel Statler)

Friday—January 29th

10:00 A. M.—Sixth Session (Ballroom, Hotel Statler)

Technical Papers:

A Study of Intermittent Operation of Oil Burners, by L. E. Seeley

Room Warming by Radiation, by A. H. Barker

A Study of the Combustible Nature of Solid Fuels, by R. V. Frost

Installation of Officers

Resolutions

Adjournment

12:30 P. M.—Luncheon and Meeting of Council

2:30 P. M.—Inspection Trip to Cleveland Industrial Plants

COMMITTEES

T. A. Weager, *General Chairman*

Finance: Walter Klie, *chairman;* W. E. Stark and H. M. Nobis.

Banquet: W. C. Kammerer, *chairman;* C. W. St. Clair and Vincent Eaton.

Reception and Registration: C. F. Eveleth, *chairman;* F. A. Kitchen and C. Gottwald.

Transportation and Inspection Trips: F. H. Morris, *chairman;* C. J. Deex and J. E. Beyer.

Entertainment: R. G. Davis, *chairman;* E. H. Pogalies and S. H. Givelber.

Publicity Committee: M. F. Rather, *chairman; Akron District*—D. E. Humphrey, and *A. S. R. E.*—D. F. Keith.

Bridge Hostesses: Mrs. Walter Klie, Mrs. T. A. Weager, Mrs. H. B. Matzen, Mrs. D. F. Keith, Mrs. C. F. Eveleth and Mrs. C. W. Colby

studies conducted at the University of Minnesota and the Research Laboratory of the Society at Pittsburgh. The topics for the afternoon session are devoted to research investigations both in the Laboratory and in the field.

On Wednesday morning the technical session will be transferred to the Little Theatre in the Cleveland Auditorium and the program will be sponsored by the A. S. H. V. E., whose members will be joined by the *A. S. R. E.* men to discuss four topics of mutual interest.

On Thursday morning at 10:00 o'clock, also in the Little Theatre, a joint session will be held with Alvin H. Baer presiding, the program being arranged by the *A. S. R. E.* and including three papers on studies and applications of cooling.

Returning to the Hotel Statler in the afternoon, the subject of ventilation will be considered in the work of two important committees and a technical paper. The work of the Committee on Testing and Rating Unit Ventilators will be reported by Chairman John Howatt. W. H. Driscoll will present the Report of the Committee on Ventilation Standards. This committee for an entire year has been gathering data from all practical and scientific sources with the idea of presenting recommendations for the use of states or municipalities who desire to have modern ventilation standards for all types of buildings. This comprehensive report has been carefully prepared by a committee representing diverse viewpoints in the field of ventilation and attempts to correlate the best available information on the subject. It is anticipated that an extensive discussion will be held because of the widespread interest in modernizing present ventilation requirements. The preliminary report, presented at the Semi-Annual Meeting, indicated that this subject is a live one.

The concluding session will be held on Friday morning in the Hotel Statler and will include discussion of several papers, after which the newly elected officers will be installed.

It should be noted that the subjects presented cover a wide field and among the most important are heat transmission, steam and hot water heating, combustion, oil burning, heating by radi-

HOTEL STATLER ROOM RATES

Every Room has Private Bath, Running Ice Water, and Radio

Type

A—Room for One
With shower bath; $3, $3.50, $4.
With shower and tub; $3.50, $4, $4.50, $5, $6, $9.

B—Room for Two (one double bed)
With shower bath; $4.50, $5, $6.
With shower and tub; $5, $6, $6.50, $7, $8, $11.

C—Room for Two (twin beds)
With shower bath; $5.50, $6, $6.50.
With shower and tub; $7, $7.50, $8, $9, $12.

D—Room for Three
With shower bath; $8.50.
With shower and tub; $9, $10.50.

E—Room for Four
With shower and tub; $12, $13.

F—Living Rooms—$7.

Method of Choosing Location of, Financing and Conducting Meetings of the Society

Resolved: That inasmuch as the Annual and Semi-Annual Meetings of the Society come under the jurisdiction of the Council, the following rules governing the handling of such meetings be adopted by the Council and published in the JOURNAL of the Society at least twice during every year, preferably just prior to each meeting:

1—The Council will select the city in which the Annual or Semi-Annual Meeting is to be held, giving due consideration to the invitations received from Chapters or members as well as to the advisability of so distributing those meetings as to make them of the greatest advantage to the general membership, and to reduce as far as possible the expense of members attending.

2—That an appropriation be made to cover the entertainment or local expenses, incurred in connection with the meeting not exceeding $500.00, the regular meeting expense to be taken care of by the General Fund of the Society in the regular way.

3—That no registration fee or compulsory obligations of any nature be imposed on members or guests.

4—That the purchase of tickets for banquets or for any other form of entertainment that may be provided be entirely voluntary.

5—That the grouping of features and the same of tickets for group features be discouraged.

6—That the raising of Funds from manufacturers of heating apparatus be discouraged.

7—That the display of samples, or of literature, advertising the product of any manufacturer in any way, shape or form, be not permitted at the booths, registration desk, or in or about the meetings.

8—That the distribution of trade papers be entirely at the discretion of the committee in charge.

9—That the local Chapter, or local members, be empowered to form a General Committee with such sub-committees as may be required to handle the details of transportation, hotel accommodations, entertainment, finance, etc., and that this General Committee be requested to confer frequently with the Council, through the Secretary of the Society, and to make frequent reports on progress in connection with the various matters being handled by them.

10—That the arrangements of elaborate and costly entertainment features be discouraged.

Adopted at Council Meeting, January 29, 1926.

ation, corrosion, acoustical problems, air conditioning and cooling, and a simplified psychometric formula.

The sessions have been arranged to permit adequate presentation of the subjects and a complete discussion by those interested.

Social Events

The entertainment program for A. S. H. V. E. members includes special features for ladies and several events which will be jointly enjoyed with *A. S. R. E.* members and ladies.

On Monday, the officers of the Society and authors of papers will meet at luncheon in the Hotel Statler at 12:30 p. m., and at 7:00 p. m. the *A. S. R. E.* Council will be hosts to the A. S. H. V. E. Council and their wives at a dinner in the Hotel Cleveland. At 8:30 p. m. members of both Societies will make an inspection of the Heating and Ventilating Exposition in the Cleveland Auditorium.

On Tuesday evening, at 8:30, an informal reception for both Societies will be held in the ballroom of the Hotel Statler and the entertainment provided is a Night in Monte Carlo.

One of the features which will be attractive to both members and ladies is a motor trip to Akron, starting from the hotel headquarters at 2:00 p. m., on Wednesday, the 27th. The trip will be made in de-luxe buses and visits will be made to the Goodyear-Zeppelin factories and hangar.

In the evening at 7:00 o'clock the Past Presidents and their wives will meet for dinner in the Hotel Statler and at 8:00 o'clock the ladies will attend a theatre party at Keith's Palace.

In the Cleveland Hotel, the *A. S. R. E.* members and guests will hold their annual banquet and dance and members of both organizations are invited to attend.

On Thursday afternoon, the ladies of both organizations will participate in a bridge party at the Chamber of Commerce at 2:30.

In the evening, at 7:00 o'clock, the annual banquet and dance of the A. S. H. V. E. will be held in the ballroom of the Hotel Statler. At this meeting a speaker of note will address the gathering and the F. Paul Anderson Medal will be awarded. The customary presentation of the Past President's Emblem will be made.

After the adjournment of the meeting on Friday, the new Council will immediately organize at a luncheon meeting, while other members and guests are making an inspection trip of Cleveland industrial plants.

Every effort has been made by the Committee on Arrangements, headed by T. A. Weager, General Chairman, to prepare an attractive and entertaining program that will bring the members to Cleveland in order to make the 38th Annual Meeting the biggest on record.

With moderate hotel rates, round-trip railroad transportation at fare and one-half, and nominal charges for the Banquet, Monte Carlo Night, and trip to Akron, it is the belief of local members that everyone who comes to Cleveland, January 25-29, will be well repaid.

It is suggested that A. S. H. V. E. members invite their friends to visit Cleveland and attend the meetings and the exposition.

President Carrier and Philip Drinker Receive John Scott Medals

Willis H. Carrier, President of the Society, and Philip Drinker of Harvard School of Public Health, recently were awarded John Scott Medal with certificate and premium of $1000 by the City of Philadelphia for inventions contributing to the welfare of mankind.

The presentation to Mr. Carrier was made on December 1 during the A. S. M. E. meeting at the Hotel Astor, New York, by Louis Heiland, Secretary of the Board of Directors of City Trusts of the City of Philadelphia and Mr. Drinker's award received jointly with Mr. L. A. Shaw of Harvard University was made during the meeting of the Franklin Institute in Philadelphia.

Mr. Carrier was awarded the medal upon the recommendation of the Advisory Committee of the Board for his "invention of processes and apparatus for air conditioning and refrigeration," and the award to Messrs. Drinker and Shaw was made for the invention of an "apparatus for the administration of artificial respiration."

The John Scott Medal carrying a premium of $1000 is awarded annually through the City of Philadelphia for an invention adding to the "comfort, welfare and happiness of mankind" and dates back to 1816 when John Scott, a chemist of Edinburgh, Scotland, bequeathed to the City of Philadelphia the sum of $4000,

the income of which was to be "laid out in premiums to be distributed among ingenious men and women who make useful inventions."

Under an Act of the General Assembly of Pennsylvania in 1869 the Fund, with other charitable funds in the care of the city, was transferred to the Board of Directors of City Trusts, which consists of twelve citizens appointed by the Courts of Common Pleas. The Mayor of the City of Philadelphia and the President of the City Council are ex-officio members of the Board.

In order that the awards may be made "for useful inventions that will be useful to mankind in the advancement of chemical, medical or any other science, or in the development of industry in any form, which are definitely accomplished and may add to the comfort, welfare and happiness of mankind," the Board has selected an Advisory Committee to make recommendations annually for the awards.

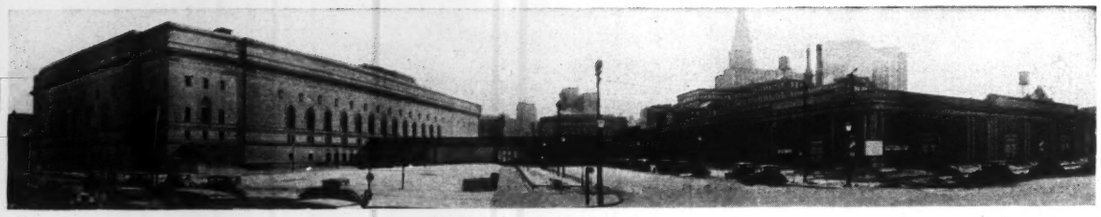
Reduced Railroad Fare for Cleveland Meeting

Members can purchase *round-trip* tickets to Cleveland at *fare and one-half* under the Identification Certificate plan from all points in the United States and Canada.

The Identification Certificates will be mailed to all members January 4, and tickets may be purchased upon the presentation of the Identification Certificate and are good for a period of 30 days and may be used via *one route* on the going trip and *another route* returning.

Tickets will be on sale in the various territories as follows:

New England States	Jan. 21-27
Eastern States	Jan. 21-27
Southeastern States	Jan. 21-27
Central States	Jan. 21-27
Southwestern States	Jan. 20-27
Western and Pacific Coast States	Jan. 17-27
Canada (Eastern)	Jan. 21-27
Canada (Western)	Jan. 19-26



CLEVELAND AUDITORIUM

Exposition Shows Trend of Heating and Ventilating

ALL branches of the heating, ventilating and air conditioning industry are eagerly awaiting the forthcoming Heating and Ventilating Exposition, to be held in Cleveland, Ohio, January 25-29, 1932, in combination with the annual meetings of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the *American Society of Refrigerating Engineers*. Preparation of the exhibits is going forward rapidly. Earlier predictions as to the array of new equipment and appliances to be shown for the first time at the exposition are borne out by the reports from exhibitors.

All of the new developments which were intended for display at the exposition will be ready for the opening. Their number and variety will furnish striking proof of the stimulating effect of the exposition on development work in this field. Knowledge that the representative men in the industry are coming to the exposition to see what is new in the way of better equipment and appliances is acting to spur on the exhibitors to fulfill their expectations.

The exposition serves not only as a market-place but as an inspiration to manufacturer and engineer alike to new achievements advancing the progress of the art.

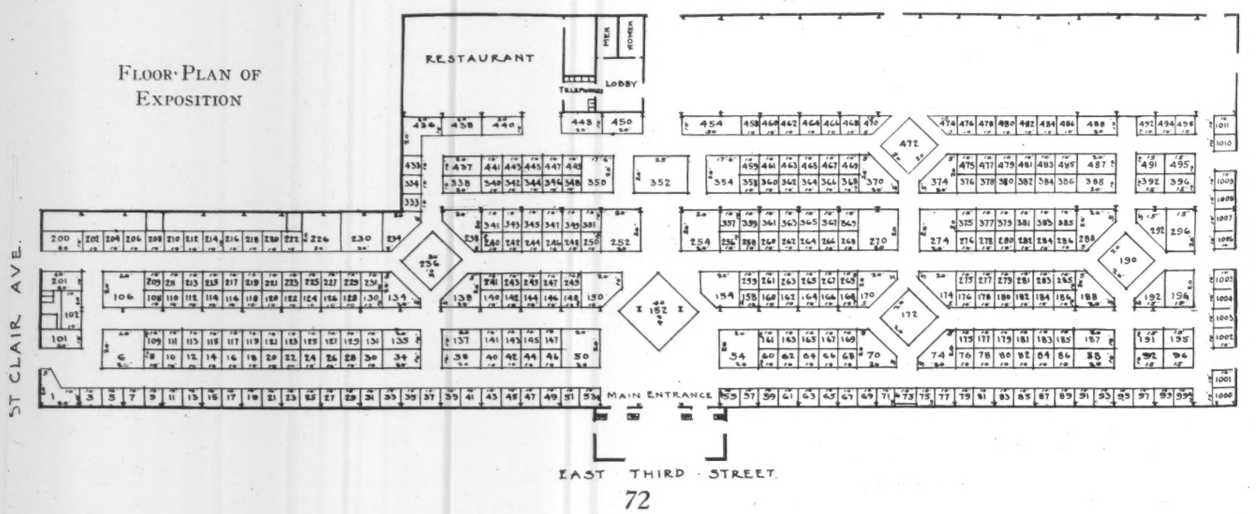
Another influence which is acting to make the exposition a rallying point for new ideas is the fact that so many homeowners and others directly interested in heating, ventilating and air conditioning for new construction and remodeling will be in attendance, looking for improved means for warming and cooling their homes. There will be a notable increase in the array of devices for house application to meet this demand.

As the opening date draws near, it is evident that the exposition will mark not only the farthest advance in heating, ventilating and air conditioning, but that it will perform a unique service in indicating future trends in the development of the art. In the ventilating field, for example, the exhibits will reflect the increasing attention which is being given to the elimination of dust and noises, particularly in offices and domestic ventilating appliances. Quiet and freedom from dust now are recognized as having an importance scarcely less than the purity of the air itself. In city offices, particularly, noises and dust, which have been tolerated so long as necessary evils, have become pressing problems as a result of the growth of automobile traffic and of the machine age generally. Several exhibits will show some unique ways in which these problems are being solved, making it certain that window ventilating systems of the future must include means for filtering the air and for silencing noises.

Another trend which will be indicated by the exposition is in connection with the heating of rooms. In several of the exhibits of baseboard radiators and concealed heating surfaces generally, the enclosures will be provided with side outlets designed to effect a horizontal air flow across the room. In the realm of warm air heating will be noted the increasing use of means for providing positive air circulation not only to the rooms themselves but to the exposed sides of the rooms from which points a low, lateral air distribution may be obtained.

The A. S. H. V. E. will have an attractive exhibit at the exposition showing working models of laboratory equipment to demonstrate how data are taken during investigations at the Research Laboratory in Pittsburgh and in the co-operating college laboratories. The working exhibits will be supplemented by pictorial groups illustrating work that has been under way in the various laboratories co-operating through the Committee on

FLOOR PLAN OF EXPOSITION



Research. This educational exhibit will demonstrate to Society members and visitors how vital research is to the progress of the profession and industry.

The list of exhibits using space is:

Space No.	
169	Adams Bros. Mfg. Co., Inc., 1500 W. North Ave., Pittsburgh, Pa.
352	Aerofin Corporation, 850 Frelinghuysen Ave., Newark, N. J.
365-367	Aerologist, The, 121 N. Clark St., Chicago, Ill.
55-57	A. G. A. Testing Laboratory, 1032 E. 62nd St., Cleveland, O.
21	Air Control Systems, Inc., 2240 N. Racine Ave., Chicago, Ill.
42-44	Airgard Ventilator Corp., 250 E. 43rd St., New York, N. Y.
1	Air Reduction Sales Co., 60 E. 42nd St., New York, N. Y.
42-44	American Air Filter Corp., 215 Central Ave., Louisville, Ky.
172	American Gas Association, 420 Lexington Ave., New York, N. Y.
275-277	American Gas Products Corp., 704 Chrysler Bldg., New York, N. Y.
190	American Oil Burner Association, 342 Madison Ave., New York, N. Y.
438	American Propeller Co., 401 Bendix Drive, South Bend, Ind.
338-340 437-441	American Radiator Co., 40 W. 40th St., New York, N. Y.
270	American Society of Heating & Ventilating Engrs., 51 Madison Ave., New York, N. Y.
102	American Society of Refrigerating Engrs., 37 W. 39th St., New York, N. Y.
369	American Temperature Indicating Co., Toledo, O.
369	American Warming & Ventilating Co., Toledo, O.
348	Ames Iron Works, Oswego, N. Y.
448-450	Ames Pump Co., 90 West St., New York, N. Y.
140	Armstrong Machine Works, Three Rivers, Mich.
476	Auer Register Co., 3608 Payne Ave., Cleveland, O.
1000-1001	Automatic Burner Corp., 1823 Carroll Ave., Chicago, Ill.
74	B-Line Boiler Co., E. 131st St. and Taft Ave., Cleveland, O.
138	Barber-Coleman Co., Rockford, Ill.
170	Barber Gas Burner Co., 3704 Superior Ave., Cleveland, O.
135	Barnes & Jones, Inc., 128 Brookside Ave., Jamaica Plain, Boston, Mass.
238	Bishop & Babcock Sales Co., 4901 Hamilton Ave., N. E. Cleveland, O.
259	Bohn Aluminum & Brass Corp., 2306 Franklin St., Detroit, Mich.
126-128	Bonney Forge & Tool Works, Allentown, Pa.
130	Borden Company, Warren, O.
43-45	Brownell Co., N. Findlay St., Dayton, O.
54	Bryant Heater & Mfg. Co., 17825 St. Clair Ave., Cleveland, O.
30-34	Buckeye Blower Co., Columbus, O.
229	Building Arts Exhibit, Inc., Builders Exchange Bldg., Cleveland, O.
165	Burdett Mfg. Co., 19 N. Sheldon Ave., Chicago, Ill.
259	Capitol Brass Division, 2306 Franklin St., Detroit, Mich.
23	Carbondale Machine Co., Carbondale, Pa.
350-352	Carrier Corporation, 850 Frelinghuysen Ave., Newark, N. J.
352	Carrier Engineering Corp., Newark N. J.
352	Carrier-Lyle Corp., 850 Frelinghuysen Ave., Newark, N. J.
350	Carrier-York Corp., 1541 Sansom St., Philadelphia, Pa.
236	W. D. Cashin Co., 69 A St., South Boston, Mass.
186	Century Electric Co., 1806 Pine St., St. Louis, Mo.
95-97	Century Engineering Corp., Cedar Rapids, Iowa
279	W. M. Chace Valve Co., 1600 Beard Ave., Detroit, Mich.
137-141	Chicago Pump Co., 2336 Wolfram St., Chicago, Ill.
	Cleveland Engineering Society, Hanna Bldg., Cleveland, O.
170	Cleveland Gas Burner & Appliance Co., Cleveland, O.
288	Cleveland Steel Products Corp., Madison at W. 74th St., Cleveland, O.
19	Cleveland Stoker Co., 6545 Carnegie Ave., Cleveland, O.
25	Coal Carburetor Co., 290 George St., New Brunswick, N. J.
63	Columbia Burner Co., Toledo, O.
480	Columbus Humidifier Co., 1 E. 5th Ave., Columbus, O.
259	Conco Temperature Control Corp., Detroit, Mich.
175	Cook Electric Co., 2700 Southport Ave., Chicago, Ill.
225-227	Cooling & Air Conditioning Corp., 11 W. 42nd St., New York, N. Y.
124	Copeland Products Co., Mt. Clemons, Mich.
142	Coppus Engineering Corp., Worcester, Mass.
46-50-147	Crane Company, 836 S. Michigan Ave., Chicago, Ill.
99A	Crystal Oil Burner Corp., 1440 Broadway, New York, N. Y.
71	N. C. Davison Gas Burner & Welding Co., 2939 Smallman St., Pittsburgh, Pa.
46	Doherty-Brehm Co., 333 N. Michigan Ave., Chicago, Ill.
274	Domestic Engineering, 1900 Prairie Ave., Chicago, Ill.
354-358-459	C. A. Dunham Co., 450 E. Ohio St., Chicago, Ill.
248-250	Struthers Dunn, Inc., 139 N. Juniper St., Philadelphia, Pa.
492	Eagle-Picher Lead Co., Joplin, Mo.
66-68	East Ohio Gas Company, 1405 E. Sixth St., Cleveland, O.
244	Economy Pumping Machinery Co., 3431 W. 48th Pl., Chicago, Ill.
376	Emerson Elec. Mfg. Co., Inc., 2012 Washington Ave., St. Louis, Mo.
274	Engineering Publications, Inc., 1900 Prairie Ave., Chicago, Ill.
164-166	Estate Stove Co., The, Hamilton, O.
366-368	Fitzgibbons Boiler Co., Inc., 570 Seventh Ave., New York, N. Y.
60-62	Forest City Foundries Co., 2500 W. 27th St., Cleveland, O.
174	Fox Furnaces Co., Elyria, Ohio
19	Fox Smith Sales Co., Cleveland, O.
204-206	Frick Company, Inc., Waynesboro, Pa.
212-214-216	Frigidaire Corp., The, Dayton, O.
187	Fuel Oil Journal, 420 Lexington Ave., New York, N. Y.
44	General Air Filters Co., Louisville, Ky.
254	General Electric Co., 1 River Road, Schenectady, N. Y.
254	General Electric Heat Regulator Co., Franklin Trust Bldg., Philadelphia, Pa.
160-162	General Gas Light Co., Kalamazoo, Mich.
1008-1009	Gilbert & Barker Mfg. Co., Springfield, Mass.
458-460	Greene Gas Cleaner Co., 1600 Union Trust Bldg., Cleveland, O.
236	Grinnell Co., Inc., Providence, R. I.
236	Grinnell Co. of Canada, Ltd., 2440 Dundas St., W. Toronto, 9, Ont., Canada
465	Hart & Cooley Mfg. Co., Holland, Mich.
292	Hart Oil Burner Corp., 2200 N. Adams St., Peoria, Ill.
192	Heating Journals, Inc., 167 Madison Ave., New York, N. Y.
274	Heating, Piping & Air Conditioning, 1900 Prairie Ave., Chicago, Ill.
333-334-433	Heating and Ventilating, 140 Lafayette St., New York, N. Y.
468-470	Henry Furnace & Foundry Co., 3471 E. 49th St., Cleveland, O.
465	Wm. Highton & Sons, Div. of Hart & Cooley Mfg. Co., Holland, Mich.
365	E. Vernon Hill Co., 121 N. Clark St., Chicago, Ill.
244-246 345-347	Hoffman Specialty Co., Inc., Waterbury, Conn.
38	Illinois Engineering Co., Racine Ave. at 21st St., Chicago, Ill.
362	Independent Air Fiter Co., Inc., 29 S. Clinton St., Chicago, Ill.
482	Independent Register & Mfg. Co., 3747 E. 93rd St., Cleveland, O.
27-29	Iron Fireman Mfg. Co., Portland, Ore.
440	Johns-Manville Corp., 292 Madison Ave., New York, N. Y.
191	S. T. Johnson Co., 940 Arlington Ave., Oakland, Calif.
361-363	Johnson Service Co., Milwaukee, Wis.
283-285	Keasbey & Mattison Co., Ambler, Pa.
231	John G. Kelly, Inc., 210 E. 45th St., New York, N. Y.
366-368	Kewanee Boiler Co., Inc., 570 Seventh Ave., New York, N. Y.
443-445	Kewanee Boiler Corporation, Kewanee, Ill.
131	Kieley & Mueller, Inc., 34 W. 13th St., New York, N. Y.
485	Kleenaire Filter Co., Stevens Point, Wis.
486	Knowles Mushroom Ventilator Co., 41 N. Moore St., New York N. Y.
464	Lakeside Company, Hermansville, Mich.
148	Leeds & Northrup Co., 4901 Stenton Ave., Philadelphia, Pa.
51-53	The Linde Air Products Co., 205 E. 42nd St., New York, N. Y.
26-28	McCord Radiator & Mfg. Co., 2587 E. Grand Blvd., Detroit, Mich.
161-163	J. H. McCormick & Co., Lewis Tower, 30th Fl., Philadelphia, Pa.
182	McDonnell & Miller, Wrigley Bldg., Chicago, Ill.
154	Jas. P. Marsh & Co., 2073 Southport Ave., Chicago, Ill.
462	Masonite Corp., 111 W. Washington St., Chicago, Ill.
238	Massachusetts Blower Co., Cleveland, O.
1002-1003 1004	May Oil Burner Corp., 3500 E. Biddle St., Baltimore, Md.
195	The Mercoid Corp., 4201 Belmont Ave., Chicago, Ill.
42	Midwest Mfg. Co., Bradford, Pa.
280	Mineral Felt Insulating Co., 2284 Albion St., Toledo, Ohio
256-258-260	Minneapolis-Honeywell Regulator Co., Minneapolis, Minn.
184	Modern Heat Regulator Co., 1781 E. 55th St., Cleveland, O.
375	Modine Mfg. Co., Racine, Wis.
92	Monarch Mfg. Works, Inc., Salmon and Westmoreland Sts., Philadelphia, Pa.
39-41	Motor Stokor Corp., 250 Park Ave., New York, N. Y.
88	Motor Wheel Corp., Lansing, Mich.
49	Mouat Vapor Heating Co., 1246 W. 4th St., Cleveland, O.
261-263	L. J. Mueller Furnace Co., 167 Reed St., Milwaukee, Wis.
179-181	Mueller Brass Co., Port Huron, Mich.
374	Multicell Radiator Corp., West Ave., Lockport, N. Y.
152	Nash Engineering Co., South Norwalk, Conn.
44	National Air Filter Co., Inc., 205 Central Ave., Louisville, Ky.
240-242 341-343	National Radiator Corp., Johnstown, Pa.
472	National Warm Air Heating Assn., 50 W. Broad St., Columbus, O.

Space No.	
454	Herman Nelson Corp., Moline, Ill.
370	Newport Boiler Div. Richardson & Boynton Co., 260 Fifth Ave., New York
202	Niagara Blower Co., 6 E. 45th St., New York, N. Y.
364	Nocare Electric Radiator Corp., 1 E. 42nd St., New York, N. Y.
223	Ohio Electric Mfg. Co., 5900 Maurice Ave., Cleveland, O.
192	Oil Heat, 167 Madison Ave., New York, N. Y.
475	Owens-Illinois Glass Co., Toledo, O.
190	Oil Heating Institute, 342 Madison Ave., New York, N. Y.
53	Oxweld Acetylene Co., 30 E. 42nd St., New York, N. Y.
357-359	Parker Appliance Co., 10320 Berea Road, Cleveland, O.
436	Parks-Cramer Co., Fitchburg, Mass.
64	Patrol Valve Co., Locust Ave. and W. 114th St., Cleveland, O.
467-469	Peerless Unit Ventilation Co., Inc., 774 Union Ave., Bridgeport Conn.
87-89	Penn Electric Switch Co., 20th and Walnut Sts., Des Moines, Iowa
168	Pennsylvania Furnace & Iron Co., Warren, Pa.
254	Penn Heat Control Corp., 20 S. 15th St., Philadelphia, Pa.
84-86	Perfection Stove Co., 7609 Platt Ave., Cleveland, O.
487-388	Petroleum Heat & Power Co., Stamford, Conn.
348-348A, 449-449A	Pierce, Butler & Pierce Mfg. Corp., 41 E. 42nd St., New York, N. Y.
241-243	Piqua Elec. Mfg. Co., Piqua, O.
282-284-286	Plumbing & Heating Contractors Trade Journal, 515 Madison Ave., New York
342	Powers Regulator Co., 2720 Greenview Ave., Chicago, Ill.
1006	Preferred Utilities Mfg. Corp., 33 W. 60th St., New York, N. Y.
234	Propellair, Inc., 17 S. High St., Columbus, O.
385	Rawlplug Co., Inc., 98 Lafayette St., New York, N. Y.
265-267-269	F. I. Raymond Co., 629 N. Washington Blvd., Chicago, Ill.
42	Reed Air Filter Co., Inc., 215 Central Ave., Louisville, Ky.
370	Richardson & Boynton Co., 260 Fifth Ave., New York
461-463	The Ric-Wil Co., 1562 Union Trust Bldg., Cleveland, O.
40	H. H. Robertson Co., Grant Bldg., Pittsburgh, Pa.
176-178	Robbins & Myers Gas Appliances, Inc., Springfield, O.
249	Rochester Circulator Sales Div., 3092 Culver Road, Rochester, N. Y.
67-69	Round Oak Furnace Co., Dowagiac, Mich.
447	W. A. Russell & Co., 5039 Grand Central Term. Bldg., New York, N. Y.
158	Ruud Manufacturing Co., 29th and Smallman Sts., Pittsburgh, Pa.
118-120	Safety Equipment Service Co., 1228 St. Clair Ave., N. E., Cleveland, O.
150	Sarco Co., Inc., 183 Madison Ave., New York, N. Y.
492	L. R. Shaw Refractories & Equipment Co., 5716 Euclid Ave., Cleveland, O.
276-278	H. M. Sheer Co., Quincy, Ill.
83-85	Silent Glow Oil Burner Corp., 1000 Park St., Hartford, Conn.
59-61	Silica-Gel Corp., Baltimore Trust Bldg., Baltimore, Md.
121-123	Skidmore Corp., St. Joseph, Mich.
35-37	Spencer Heater Co., Williamsport, Pa.
49	Stay-Rite Co., Inc., 1250 W. Fourth St., Cleveland, O.
47	Sun Radiator Cover, Inc., 7802 Carnegie Ave., Cleveland, O.
70	Surface Combustion Corp., 2375 Dorr St., Toledo, O.
33	Swartwout Co., 18511 Euclid Ave., Cleveland, O.
143-145	Taco Heaters, Inc., 342 Madison Ave., New York, N. Y.
127	Webster Tallmadge & Co., Inc., 50 Church St., New York, N. Y.
256	Time-O-Stat Control Co., Elkhart, Ind.
188	Timken Silent Automatic Co., 100-400 Clark Ave., Detroit, Mich.
268	H. O. Trerice Co., 1420 W. Lafayette St., Detroit, Mich.
486	Tuttle & Bailey Mfg. Co., 155 E. 44th St., New York, N. Y.
477	Uni-Flo Grille Products Co., 2265 W. Grand Blvd., Detroit, Mich.
51	Union Carbide Co., 205 E. 42nd St., New York, N. Y.
485	United States Hame Co., 135 Tonawanda St., Buffalo, N. Y.
144-146, 245-247	United States Radiator Corp., Detroit, Mich.
129	Voss, Inc., J. H. H., 408 Concord Ave., New York, N. Y.
386	Wagner Electric Corp., 6400 Plymouth Ave., St. Louis, Mo.
252	Warren Webster & Co., Camden, N. J.
360	Warm Air Furnace Fan Co., 6545 Carnegie Ave., Cleveland, O.
474	Warm-Air Heating, 105 S. 9th St., St. Louis, Mo.
231	Watts Regulator Co., Lawrence, Mass.
262-264-266	Weil-McLain Co., 641 W. Lake St., Chicago, Ill.
134	W. P. Whittington, Inc., 122 E. Michigan St., Indianapolis, Ind.
466	L. J. Wing Mfg. Co., 154 W. 14th St., New York, N. Y.
226	York Ice Machinery Corp., York, Pa.
91-93	York Oil Burner Co., York, Pa.
167	Young Radiator Co., Racine, Wis.
344-346	Youngstown Sheet & Tube Co., Youngstown, Ohio

Death of Edward Addy

It is with deep regret that the Ontario Chapter announces the loss of their member and friend, Edward Addy, who died December 2 in his sixty-second year.

He was born February 7, 1870, in County Covan, Ireland, and after completing his schooling was apprenticed in heating and plumbing work in 1888. Commencing in 1910 and for a number of years thereafter Mr. Addy was identified with the Dominion Radiator Co., Toronto, as a sales and service engineer and specialized in the design and installation of heating and ventilating systems for all classes of industrial and public buildings. For several years his work with Warden King Co. of Toronto was of similar type.

At the time of his death Mr. Addy was supervising engineer for the Board of Education, Toronto, and had charge of all heating and ventilating and plumbing installations.

Mr. Addy was elected to membership in the Society in 1923 and had taken an active interest in the work of the Ontario Chapter where he had many good friends.

His funeral was held from the Fairlawn United Church, Toronto, December 5.

The Officers and Council of the Society join with the Toronto members in extending to Mr. Addy's family and associates an expression of their deep sorrow for the loss that they have sustained by his death.

Death of John Gassler

It is with sorrow that members will learn of the death of John H. Gassler, of Jacksonville, Fla., who passed away November 2, 1931. Mr. Gassler was born June 23, 1903, in New York City and prepared for his engineering work at Mechanics Institute and Pratt Institute, Brooklyn.

In January, 1923, he joined the engineering staff of Crane Co. at Jacksonville, Fla., and at the time of his death was manager of the department.

Mr. Gassler specialized in heating work, particularly in design and construction, and was elected to membership in the Society as a Junior in 1927 and was advanced to Member grade in July 1931. He also held membership in the Florida Engineering Society and the Jacksonville Engineers Club.

His associates and many friends will learn with regret of his untimely passing and join with the Officers and Council of the Society in expressing their sorrow to his widow who survives.

Furnace Men Discuss Mechanical Systems of Heating

AT THE 19th annual convention of the *National Warm Air Heating Association*, held in the Mayflower Hotel, Washington, D. C., December 3 to 5, 1931, the subjects of air conditioning in the home and mechanical systems of warm air heating were the featured topics of discussion.

At the opening session, President I. L. Jones presided and introduced Prof. F. B. Rowley, first vice-president of the A.S.H.V.E. and Director of the Engineering Experimental Laboratory University of Minnesota, who addressed the members on the subject of Air Conditioning in the Home. A survey of the field of air conditioning was given by Professor Rowley who also discussed the three kinds of temperature considered in air conditioning and cooling work. He described the present three methods of cooling by the use of cold water, ice and mechanical refrigeration and pointed out some of the problems confronting the designer in meeting summer requirements.

Prof. A. C. Willard, University of Illinois, spoke on the subject, Our Coming Research in Mechanical Warm Air Heating. This interesting talk was illustrated by lantern slides. It was explained that the object of making the proposed installation of the all mechanical system of warm air heating at the Research Residence in Urbana, was to determine both performance and operating data which will be of general interest and assistance to the members in designing, installing and operating complete systems of heating, ventilating and air conditioning for residences.

Following the session, the members were received at the White House by President Herbert Hoover.

The speaker at the opening of the afternoon session was Hugh L. Dryden of the U. S. Bureau of Standards, who talked on the Relationship of the Bureau to the Problems of Heating and Ventilating. Following this address, F. C. Sedgwick reported for the Committee on Research; Prof. J. D. Hoffman, Chairman of the Code Committee, reported of the Committee's readiness to take up the problem of a Code for Forced Air Heating Systems and F. J. Mehrings explained the activities of the Trade Relations Committee. A. W. Wreiden told of the work of the Public Information Committee and Ralph Blanchard reported on the results obtained by the Membership Committee. Allen W. Williams, Managing Director, gave his report as Secretary and Treasurer.

The Nominating Committee presented the names of the following officers for re-election: Irving L. Jones, *President;* W. L. McGrath, *First Vice-President;* H. T. Richardson, *Second Vice-President;* Allen W. Williams, *Secretary-Treasurer; Directors,* two years: H. P. Mueller, R. W. Blanchard, F. E. Mehrings.

On Thursday evening, the annual banquet was held with H. T. Richardson acting as master of ceremonies. A warm air furnace presented by Washington Irving to Christ Church, Tarrytown, N. Y. in 1836 and used continuously for 75 years, was presented to the Association by D. Rait Richardson. President Jones acknowledged the gift to the Association and then A. P. Lamneck, Congressman from Ohio, presented the furnace to Dr. Wetmore of the Smithsonian Institution. The guest speaker, Captain Eugene Gempel, Chemical Warfare Service, gave an interesting discussion of the work of this branch of the service.

Vice-President W. L. McGrath opened the Friday morning session and introduced Dr. L. R. Thompson, Assistant Surgeon General, U. S. Public Health Service, who spoke on the subject of Good Air and Health.

The subject of Merchandising was discussed by W. H. Kratzer, Detroit, and this talk developed in an extensive and entertaining discussion.

At the afternoon session the opening address was made by Dr. S. I. Miller, New York, whose subject was Credit Granting. The concluding presentation was entitled, Latest Developments from Our Research Work by Prof. A. P. Kratz and S. Konzo of the University of Illinois, Urbana.

Engineering Societies Study Cooperative Plan

At a conference of members representing various associations at the Engineering Societies Building, New York City, on November 6, it was thought desirable, by those present, that an Engineering Committee on Applied Science be created for the purpose of disseminating information on applied science, which is common to most branches of engineering.

W. A. Shoudy stated that he would bring the matter before the Committee on Professional Divisions of the *A. S. M. E.* to ascertain its interest in sponsorship of this project in cooperation with other Societies. The opinion was expressed that one of the present disadvantages of pooling engineering information was the inability to get discussions from all interested fields on a subject broadly treated in a technical paper and that this might be overcome through the activities of the committee suggested.

Those in attendance were W. H. Carrier, President, A. S. H. V. E.; Dr. C. R. Downs, *A. I. C. E.;* Thornton Lewis, A. S. H. V. E.; Glenn Muffly, *A. S. R. E.;* and W. A. Shoudy, *A. S. M. E.*

Codes for Members

Copies of four of the Society's Standard Codes have been printed and mailed to members during December 1931. In addition to these codes, a valuable pamphlet is included which gives an Index to the complete Transactions of the Society from 1895-1930 inclusive.

MEMBERS OF NATIONAL WARM AIR HEATING ASSOCIATION AT WHITE HOUSE RECEPTION, DEC. 3, 1931

PRESIDENT HOOVER OPENS CONFERENCE ON HOME BUILDING AND HOME OWNERSHIP, DEC. 2, 1931, IN CONSTITUTION HALL, WASHINGTON, D. C.

President's Conference on Home Building and Home Ownership

SEVERAL thousand people responded to an invitation to attend the President's Conference on Home Building and Home Ownership called by President Herbert Hoover in Washington, D. C., December 2 to 5, 1931, where the results of the comprehensive study of every phase of home ownership were presented for discussion.

Registration of the thousands of visitors was held at the new Department of Commerce Building, 400 14th St., N.W., the conference headquarters.

At 8:30 p. m. on Wednesday evening, the conference was opened in Constitution Hall when President Hoover gave a brief address on the Objectives of the Conference and summarized his ideas of the value and necessity of home ownership by pointing out that, "adequate housing goes to the very roots of the well-being of the family, as the family is the social unit of the nation. It is more than comfort that is involved, it has important aspects of health and morals and education and the provision of a fair chance for growing childhood. Nothing contributes more to happiness or sound social stability than the surroundings of their homes. It should be possible in our country for anybody of sound character and industrious habits to provide himself with adequate housing and preferably to buy his own home."

The Planning Committee of 32 members consisting of architects, contractors, bankers, engineers and representatives of women's clubs, government bureaus and civic organizations under the co-chairmanship of the Secretaries of Commerce and Interior, the Hon. R. P. Lamont and the Hon. R. Lyman Wilbur, had set up 25 fact-finding committees and six correlating committees to prepare all available data on construction and housing. The report of the Committee on Fundamental Equipment, Collins P. Bliss, Chairman, covered the subjects of heating, ventilating, air conditioning, refrigeration, electric wiring, plumbing and sanitation and gave a complete summary of present day knowledge and practice in these fields.

The report of 141 pages with additional appendices is divided into four sections: (1) Heating, ventilating and air conditioning was prepared by A. C. Willard, Philip Drinker, W. H. Driscoll and Perry West; (2) Plumbing and sanitation was prepared by A. E. Hansen, E. F. Carter, W. H. Driscoll and J. L. Murphy; (3) Electric wiring and lighting, was prepared by Dr. M. Luckiesh, E. W. Commery, R. T. Jones, M. L. Nichols and R. A. Palmer; (4) Refrigeration, was prepared by Dr. Louise Stanley, G. B. Bright, Dr. Jesse Churchill, Dr. Mary Pennington and Mrs. Paul Howe.

The report of the committee was made on Thursday, December 3, 1931 at a meeting in the Chamber of Commerce of the United States by Collins P. Bliss, Chairman, who said, "that the Committee has been chiefly concerned with how to make a *home* out of every *house*." The report was presented for the guidance of people of moderate means, who desire to invest from $2,000 to $10,000 in a home. The various sub-committee chairmen presented their sections of the report and comments and suggestions of those in attendance were discussed by the committee members.

During the several days of the conference each of the 25 committees reported at the numerous meeting places where the sessions were held.

On Friday evening, President Hoover and Mrs. Hoover received the members of the conference at the White House and in the evening at 8:30 a joint session of the conference was held in Constitution Hall with Hon. R. P. Lamont presiding and Mrs. Jane D. Rippin and the Hon. R. Lyman Wilbur as the speakers.

On Saturday morning the various correlating committees held their meetings and that on Technological Development, Dr. George K. Burgess, Chairman, held at the Hotel Washington, covered the various developments in construction and equipment. The reports given were as follows:

Brief Summary of the Report—H. L. Whittemore.
Electricity in the Home of the Future—Willis R. Whitney.
Labor Saving Tools for House Construction—Collins P. Bliss.
Teaching Students the Fundamentals of Good House Construction—Clarence A. Martin.
The Shop-Fabricated House—T. J. Foster.
Welding Operations for the Small Home—H. H. Moss.
Fire Safety in the Home—C. T. Bissell.
Air Conditioning for the Small Home—W. H. Carrier.
Technical Developments in the Use of Paint for the Home—Henry A. Gardner.

Local Chapter Reports

Cleveland

November 4, 1931. Pres. W. E. Stark called the meeting of the Cleveland Chapter to order and a communication regarding unemployment from A. V. Hutchinson, National Secretary, was read. After some discussion, a motion was carried to have an Employment Officer selected by the Board of Governors.

The Chapter elected two new members, R. G. Olson and M. B. Wright.

President Stark spoke of the value of membership in the Society and stressed the importance of obtaining a great number of prospective members. He also discussed the idea of Limited Chapter Membership and it was moved that the Board of Governors work out such a plan, giving consideration to age limit.

T. A. Weager, general chairman of the Committee in charge of the Society's 38th Annual Meeting to be held in Cleveland, January 25-29, 1932, spoke of some of the arrangements which have been made to provide for the entertainment and welfare of the members and guests during their visit.

The Chapter was privileged to have present G. Harold Noyes, Senior Meteorologist of the Cleveland Weather Bureau, who spoke on Some Aspects of the Weather in Regard to Heating and Ventilating. Mr. Noyes discussed the factors affecting forecasting of the weather, routine of the Weather Bureau, and problems that develop in making forecasts. He pointed out that it was necessary to get the readings of approximately 100 barometers within a wide radius of Cleveland, weather conditions, wind velocities, wind directions, etc., before the Weather Bureau could make forecasts that were correct about eight times out of ten. Mr. Noyes told of the work being done in Cleveland to obtain information by sending up an airplane to high altitudes equipped with self-recording instruments. The speaker also had considerable general information regarding the weather and its relation to heating and ventilating.

An interesting discussion followed the talk, at the conclusion of which, Mr. Noyes was given a vote of thanks.

Kansas City

November 9, 1931. Fifteen members of the Kansas City Chapter gathered at the Ambassador Hotel to attend a dinner and meeting.

Vice-Pres. W. A. Russell called the meeting to order and a statement of the activities of the A. S. H. V. E. from the national headquarters was read by D. D. Zink, Secretary, and discussed by the members. Particular attention was given to membership and employment activities of the Chapter.

The principal speaker of the evening was E. N. McDonnell, McDonnell and Miller, Chicago, who discussed the application of water feeders as well as their design and operation. Mr. McDonnell also gave the members some idea on boiler operation and causes of failure. An animated discussion followed the conclusion of his talk.

Prof. A. H. Sluss of the University of Kansas told of his efforts on the Governor's Coal Committee of Kansas and indicated that it was his intention to hold a class at Kansas University on Coal Utilization at an early date.

Illinois

November 9, 1931. In the Cameo Room of the Hotel Morrison, 125 members and guests of the Illinois Chapter gathered for a meeting, 80 of whom had been present earlier in the evening at dinner. The absence of Pres. J. J. Aeberly was explained by Vice-Pres. J. H. O'Brien, who presided.

The appointment of A. G. Sutcliffe as Employment Representative of the Illinois Chapter was announced, following the reading of a communication from the headquarters office regarding an employment service plan.

The revised form of the Constitution and By-Laws, as prepared by the committee, was presented by M. S. Good, and it was proposed that this be submitted to the members in the manner required by the Constitution and By-Laws now in effect, but no vote was taken at this time.

H. B. Johns, of the House Heating Division, Peoples Gas, Light and Coke Co., was introduced by John Howatt, and showed a three reel film, picturing the bringing of natural gas to Chicago. Mr. Johns was assisted by Ernest Beaumont and Edward J. Hanlon. A spirited discussion followed the presentation of the film, in which Mr. Johns answered the questions put to him in an able and thoroughly convincing manner.

The Secretary was instructed to write to the Peoples Gas, Light and Coke Co., and congratulate them on their tremendous achievement and thank them for having Mr. Johns and the other gentlemen address the meeting. The Chapter expressed their appreciation for it was one of the most interesting and instructive talks ever heard before the Chapter.

Massachusetts

December 7, 1931. Fifty-two members and friends of the Massachusetts Chapter enjoyed a dinner in the Boston Chamber of Commerce Building and later the attendance was greatly increased by members who came to hear W. H. Carrier, President of the Society, whose subject was Servicing the Human Power Plant.

President Carrier gave an illustrated lecture which dealt with the heat generated and given off by the human body and the comfort zone for a ventilated enclosure in varying seasons. In concluding, President Carrier asked for discussion from Philip Drinker and C. P. Yaglou of the Harvard School of Public Health and others.

A. V. Hutchinson, National Secretary, spoke briefly of the activities of the National Society, particularly outlining the program of the Annual Meeting to be held in Cleveland.

C. P. Yaglou was elected to serve on the Nominating Committee for the coming year as the representative of the Massachusetts Chapter with J. W. Brinton as alternate.

Western New York

November 9, 1931. At a dinner and meeting held in the Hotel Buffalo, the Western New York Chapter was host to the Officers and Council of the Society, who earlier in the day had attended a luncheon at the Hotel Statler with the Chapter Officers and Board of Governors. Pres. Joseph Davis presided over the meeting at which 117 members and guests were present and announced the appointment of J. J. Yager as Chapter Employment Officer and L. A. Harding as chairman of the Nominating Committee.

Pres. W. H. Carrier was introduced as the speaker of the evening and gave a very interesting and well illustrated talk on Servicing the Human Power Plant. Later he discussed the efforts of the Society's Committee on Ventilation Standards to prepare a Ventilation Code for Legislative Purposes and asked for comments. An open discussion followed, participated in by J. J. Aeberly, Chicago, W. H. Driscoll, New York, A. C. Willard, Urbana, Ill., J. R. McColl, Detroit, F. Paul Anderson, Lexington, Ky., C. L. Riley, New York, W. A. Rowe, Detroit, L. A. Harding, Buffalo, and Hugo Hutzel. It was the consensus

of opinion of those present that the Code should be adopted by the Society at the Cleveland Meeting and the committee be continued to make amendments as experience demands.

The members of the Chapter express their appreciation of the visit of the National Officers and Council Members and the very interesting address given by President Carrier. A better understanding of the National Society activities has resulted from contact with these men who are giving so unselfishly of their time and knowledge to further the art and industry.

Ontario

November 10, 1931. A special meeting of the Ontario Chapter was called on Tuesday evening in the Empire Room of the Prince George Hotel. Seventy-eight members and guests were present to greet W. H. Carrier, President of the A. S. H. V. E.

F. Frank Dyer, the winner of the Toronto Chapter's University of Toronto Award for the best thesis on heating and ventilation, was introduced to the Chapter by M. F. Thomas. Mr. Thomas extended to Mr. Dyer the Chapter's congratulations for his thesis, entitled Comparison of Warm Air and Hot Water Heating for Small Homes.

A. V. Hutchinson, Secretary of the Society, spoke to the Chapter of the meetings of the Council and Committee on Research, which had been held in Buffalo just prior to the Ontario Chapter Meeting. Mr. Hutchinson also explained arrangements for the Society's Annual Meeting in Cleveland, which will be held January 25-29, 1932, at the same time as the Annual Meeting of the *American Society of Refrigerating Engineers* and the Second International Heating and Ventilating Exposition.

E. Holt Gurney, a member of the Council, introduced the principal speaker of the evening, President Carrier. Mr. Carrier gave an illustrated talk on the subject of Economic Aspect of Air Conditioning for Human Comfort. This address was one of the most interesting and instructive talks that the Chapter had ever heard and a hearty vote of thanks was given to President Carrier by all of those present.

Pacific Northwest

November 5, 1931. After the usual dinner, the meeting was called to order by President W. E. Beggs at 7:30 p. m. in the Pine Tree Tea Room of the Shopping Tower.

The special committee appointed at the preceding meeting to study the Proposed Ventilation Code stated that the Committee had made a report to the National Headquarters and had received a reply advising that the report would be considered in the final drafting of the Code.

S. C. Miller of the Elliott Co. outlined briefly the conditions governing the selection of Prime Movers.

Prof. E. O. Eastwood reported on the Research activities of the Society during the year 1931.

Professor Eastwood also extended an invitation to the Society through the University Student Body on the Home-Coming Event to attend their banquet at the University on November 11.

The minutes of the preceding meeting were read and approved and as there was no further business, the meeting was adjourned.

September 24, 1931. A dinner and Chapter Meeting was held in the Shopping Tower Tea Room with Pres. W. E. Beggs presiding.

After the minutes of the previous meeting had been read and approved, Mr. Pollard of the Puget Sound Power & Light Co. presented some interesting data on the company's steam heating plants and steam and water electric power generation plants.

A picture illustrating the manufacture of wrought iron pipe was shown by Mr. Montgomery of the Northwest Plumbing Co.

It was voted that the chairman appoint a committee of three to study the Proposed Ventilation Code and make a report to the Society's Committee on Ventilation Standards, and that this committee have full power to report its findings in the name of the Chapter. The members appointed were W. L. Dudley, E. O. Eastwood and C. G. Zokelt.

Philadelphia

November 12, 1931. The meeting of the Philadelphia Chapter was held in the Engineers Club with Pres. E. N. Sanbern presiding. On this occasion several papers were presented on Better Cooperation Between Architects, Engineers and Contractors.

The first speaker of the evening was John H. Rankin, Past President of the Pennsylvania Section, *American Institute of Architects,* who outlined briefly just what architects expect from engineers and contractors. Mr. Rankin stated it was his opinion that the registry of engineers and architects should be restricted to those thoroughly educated in their particular field, that the engineer designing the structural portion of a building should confine his specifications to that part of the work, and that heating and ventilating engineering should be carried on by a man thoroughly familiar with that design.

J. W. Pisel, representing the engineers, spoke generally of the relations between the architect and the engineer, and between the consulting engineers and the contractor. Mr. Pisel emphasized the point that the consulting engineer should specify the equipment which he believes would be best adapted to the type of building in question and should insist that the contractor furnish the equipment without any omissions or substitutions.

The third speaker was W. F. Smith, representing the contractors. Mr. Smith pointed out the difficulties that are sometimes encountered by the contractor in endeavoring to correct mistakes made by both the architect and the engineer.

Ralph B. Bencker, President of the Philadelphia Chapter, *American Institute of Architects,* was of the opinion that the architect should prepare the architectural plans, working in harmony with the consulting engineer, and that the contractor should then assume responsibility for furnishing exactly what the specifications called for, turning the building over to the owners as a first class finished project.

The next speaker was A. E. Dambly who represented the consulting engineer. Mr. Dambly mentioned the difficulties that are encountered by the engineer when he is called in after the architectural plans have progressed to a certain point. He stated that the consulting engineer and the architect should work together more closely from the very beginning on the design and plan of the building.

The sixth speaker was A. C. Edgar, representing the contractor, who reviewed generally the relationship between architects, engineers and contractors, and made a short comparison between conditions as they exist today and 35 or 40 years ago.

At the conclusion of the papers, comments and suggestions were presented and a rising vote of thanks was extended to the speakers for their cooperation in making it one of the best and instructive meetings held in recent years.

The Nominating Committee presented the following names for Chapter Officers for 1932: *President,* L. C. Davidson; *Vice-President,* M. F. Blankin; *Secretary,* W. P. Culbert; *Treasurer,* J. H. Hucker; *Board of Governors,* W. R. Eichberg, H. F. Rettew, and E. N. Sanbern.

A motion was made and carried that the Secretary be instructed to write a letter of sympathy to H. G. Black upon the death of his mother.

There being no further business, the meeting was adjourned.

Pittsburgh

December 7, 1931. The annual meeting of the Chapter was held at McCann's Dining Rooms, Pittsburgh, where President Houghten called the meeting to order with an attendance of 50 members and guests.

Following the reading of the minutes of the previous meeting, which were approved as read, H. B. Orr, treasurer, presented his report, stating that a third of the members had not paid

their local dues for the year 1931 and he appealed to the delinquent members to pay promptly.

President Houghten then presented two out-of-town guests and members of the Society, Cliff Presdee, Chicago, and S. E. Larson, West Orange, N. J. Mr. Presdee brought greetings from the Illinois Chapter, and complimented the Pittsburgh Chapter on its meeting attendance.

G. S. McEllroy was appointed by President Houghten to serve on the Committee on Unemployment, established by the Society.

The nominations for the officers for the coming year were then presented, and after the motion was duly moved and seconded the secretary announced that the following would hold office for the year 1932:

President—R. B. Stanger
Vice-President—F. C. McIntosh
Secretary—J. L. Blackshaw
Treasurer—H. B. Orr
Board of Governors—F. C. Houghten, G. S. McEllroy and H. Lee Moore.

President-elect Stanger addressed the meeting, expressing the hope that despite the depression the new year would see a strengthened Chapter working for the good of the profession and for the Society. He also stated that any society can be only as good as its members make it.

The principal speakers of the evening was J. R. Hertzler of New York, manager of the air conditioning department of the York Ice Machinery Corp., who delivered an interesting and instructive illustrated talk on Refrigeration as Applied to Air Conditioning. Mr. Hertzler told some of the early history of air conditioning, which began with the work of Joseph McCreary, who in 1896 planned to refresh air by passing it past plates over which water flowed. By slides he illustrated many of the bulky early types of refrigerating machines and compared them with the compact modern machines of identical capacities. He showed the arrangement and operation of a typical air conditioning system, and explained different types of refrigeration plants, using ammonia, CO_2, and F12 cycles. The component parts of refrigerating plants—compressors, evaporators and condensers—were illustrated and discussed in detail. In concluding his talk, Mr. Hertzler summed up the factors which must be considered in the selection of a refrigerator, which are required load, load factor, basis of fixed charges, hours system is to be used, operating charge, flexibility of operation, space utilized, dependability, safety and noise.

After an interesting discussion a rising vote of thanks was tendered Mr. Hertzler by the members and guests in appreciation of his talk, which was followed by adjournment.

November 9, 1931. There were 65 members and guests present when Pres. F. C. Houghten called the meeting of the Pittsburgh Chapter to order in McCann's Private Dining Room.

W. W. Stevenson, chairman of the Nominating Committee, reported the following nominations for officers for the year 1932: *President,* R. B. Stanger; *Vice-President,* F. C. McIntosh; *Secretary,* J. L. Blackshaw; *Treasurer,* H. B. Orr; *Board of Governors,* F. C. Houghten, G. S. McEllroy, H. Lee Moore.

F. C. McIntosh was appointed by President Houghten to represent the Chapter on the National Nominating Committee, with T. M. Dugan as alternate.

President Houghten brought to the attention of the meeting the fact that two of the Chapter members are presidents of national engineering organizations. He first presented Mr. Stevenson, President of the *National District Heating Association,* who invited the members and guests to attend the annual convention of his association to be held in Pittsburgh in June. Mr. Dugan, President of the *American Society of Sanitary Engineers,* was then introduced and extended an invitation to the national convention to be held in Rochester, N. Y., in August.

The Secretary read a letter from the Headquarters Office of the A. S. H. V. E. which gave a statement of the Society's membership and told of its activities, particularly the work which it is doing on unemployment.

The meeting was addressed by Dwight L. McNulty, mechanical engineer for the Byllesby Engineering and Management Corp., who gave a most interesting and instructive talk on The Design and Construction of Underground Steam Lines, with particular reference to some problems encountered in Pittsburgh. The paper gave the details of the construction of Pittsburgh's large central district heating system and steam generating plant. Mr. McNulty discussed the advantages of such a system and illustrated his talk with photographs and plans of the work.

A rising vote of thanks was given to Mr. McNulty by the members and guests in appreciation of his address.

St. Louis

December 2, 1931. The December Meeting of the St. Louis Chapter was held at the Roosevelt Hotel and was called to order by Pres. R. M. Rosebrough.

L. Walter Moon, chairman of the Nominating Committee, reported the nominations and the ticket as submitted was unanimously elected, so that the following men now hold office:

President—James M. Foster.
1st Vice-President—Clinton A. Pickett.
2nd Vice-President—Paul W. Sodemann.
Secretary—C. R. Davis.
Treasurer—A. L. Walters.
Board of Governors—R. M. Rosebrough, E. A. White, G. W. F. Myers, and L. W. Moon.

Mr. Rosebrough, retiring president, thanked the members of the various committees for their cooperation and support during his administration.

President Foster spoke enthusiastically on building up the Chapter and stimulating interest in the Society.

The Membership Committee, reporting through Mr. Myers, presented the applications of 6 Limited Chapter Members.

A motion was made by Mr. Myers that a Publicity Committee be appointed for 1932 in addition to the usual committees and this proposal met with general acceptance.

President Foster appointed an Auditing Committee consisting of Messrs. Moon, McMorran and Sodemann with instructions to audit the books of the present administration and report at the next Chapter Meeting.

November 18, 1931. The St. Louis Chapter held a dinner meeting at the Roosevelt Hotel with 11 members and 10 guests present. Pres. R. M. Rosebrough called the meeting to order.

A Nominating Committee, consisting of 5 members, was appointed with L. Walter Moon, chairman, to select candidates for officers of the St. Louis Chapter for the year 1932.

After the business meeting, Mr. Moon, chairman of the Entertainment Committee, introduced E. F. Edwards and L. J. DuBois of the York Ice Machinery Corp., York, Pa. Mr. Edwards discussed refrigeration as applied to air conditioning and comfort cooling and stressed the tremendous possibilities of this phase of work in homes.

Southern California

December 8, 1931. A dinner meeting followed by an inspection of a motor manufacturing plant was enjoyed by 37 members of the Southern California Chapter at the regular monthly meeting.

Pres. Franklin Winch presided over the meeting and, after committee reports and matters concerning the Chapter had been disposed of, he introduced R. D. Ulrey, of the U. S. Electrical Co., who gave a description of the general principles of motor manufacture with particular reference to quiet operating motors for ventilation work.

After a very interesting and enlightening discussion between the members of the chapter and guests who were familiar with the subject, the assemblage made an inspection trip through the company's plant, where all machinery was in operation and permitted an excellent opportunity to see the manufacture of a motor from start to finish.

Kansas City

December 14, 1931. An unusually interesting meeting was called by W. A. Russell, vice-president, at the Ambassador Hotel. Forty members and guests were present to hear and participate in a discussion of fundamental ventilation principles.

Among the visitors were George Melcher and L. H. Brotherson of the technical staff of the Kansas City School Boards.

The routine business included the reading of a preliminary statement of the activities planned for the Heating and Ventilating Exposition at Cleveland, and an invitation to participate. The regular reports, the Treasurer's report being so good that discussion regarding reduction of dues, followed. It was decided that inasmuch as many members have paid 1932 dues that any change would be postponed for a year. It was decided to put several hundred dollars in a savings account to accumulate interest. Benjamin Natkin was appointed to serve on the National Nominating Committee.

The guest speakers were E. H. Beling and L. F. Nordine of Moline, Ill. Mr. Nordine read a very interesting paper on Ventilation which was followed by a lively discussion. M. A. Campbell, Sydney Pines, Nate Downes, Messrs. Millis, Kitchen and Brotherson discussed different phases of the paper and technical questions were answered by Mr. Beling.

A very interesting analysis of the problem of ventilation was contributed to by nearly the entire membership and everyone felt that the evening was well and profitably spent.

Michigan

December 14, 1931. About 50 members and guests were present at the December Meeting of the Michigan Chapter of the American Society of Heating and Ventilating Engineers. After dinner the minutes of the November meeting were read. President Giguere nominated Mr. Winans as chapter representative on the Nominating Committee with W. G. Boales as alternate. Mr. Boales gave the cost to attend the winter meeting which will be held in Cleveland in January.

The principal speaker of the evening was A. C. Wallich of the Wallich Ice Machine Co., who gave a very interesting and instructive lecture on refrigerating machinery. His lecture was illustrated by lantern slides. Mr. Wallich brought to the attention of those present the penalties involved in poor design of a refrigerating system. There was considerable discussion from the floor ranging from large air conditioning jobs to household air conditioning. A very considerable interest was shown in the air conditioning of homes.

Mr. Dubry made a motion to invite the National Society to hold its summer meeting in Detroit or the vicinity of Detroit in 1933.

After expressing the Society's thanks to Mr. Wallich, the meeting was adjourned.

New York

December 21, 1931. On Monday evening, December 21, 40 members of the New York Chapter assembled at the Building Trades Club, 2 Park Ave., New York, to enjoy a holiday dinner and a talk by Harold L. Alt on his experiences in foreign countries.

The meeting was opened by Pres. Russell Donnelly and the minutes of the previous meeting were read by Secretary Swain and approved.

The speaker of the evening was then introduced and gave a very interesting talk on his experiences doing engineering work, particularly in China and Russia.

He had a great many slides which served to illustrate his interesting and frequently humorous references about specific experiences both in travelling and on the job.

The audience enjoyed his explanation of native customs and experiences with Chinese and Russian mechanics and laborers, all of which served to amuse and frequently annoy American engineers, who undertook this service in foreign countries, where language, food, money and ideas were unfamiliar.

The members expressed their appreciation of Mr. Alt's efforts in bringing to them such an intimate picture of what the American engineer finds in foreign lands.

Among the guests, who enjoyed meeting many old friends, was J. A. Donnelly of Largent, W. Va., who formerly took an active part in New York Chapter affairs.

Pacific Northwest

November 28, 1931. Pres. W. E. Beggs called the meeting to order at 1:30 p. m. and brought two communications from the Headquarters Office to the attention of the Chapter. The first notice presented the Society's plan to aid its members in securing employment and the second was in regard to the building up of the Society by paying the dues promptly and securing new members. President Beggs remarked that the A. S. H. V. E. is the only Society devoted to the promotion of the science of heating and ventilating.

The meeting was adjourned and 25 members and guests made an inspection trip through the new Ford Motor Plant, where they saw the longest pipe coil in the world and many other interesting items pertaining to the mechanical equipment.

CANDIDATES FOR MEMBERSHIP

The Constitution of the Society, as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the names of applicants and their references shall be printed in the next issue of the Journal of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Membership Committee as soon as possible.

When the Membership Committee has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 24 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Membership Committee, and in turn the Council, urge the members to assume their share of the responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some members by Jan. 15, 1932, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary immediately after election.

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
AMUNDSON, LELAND ROMAINE, Research Fellow, Heating & Ventilation, University of Minnesota, Minneapolis, Minn.	F. B. Rowley A. B. Algren	M. S. Wunderlich Albert Buenger
CARLSON, EVERETT E., Br. Mgr., The Powers Regulator Co., St. Louis, Mo. (*Advancement*)	J. W. Cooper G. W. F. Myers	J. M. Foster C. A. Pickett
CLODFELTER, JOHN LUTHER, Supt., Carolina Sheet Metal Corp., Philadelphia, Pa.	W. R. Eichberg W. R. Murphy	W. P. Culbert H. G. Black
EAGLETON, STERLING PAUL, Sales Engr., Carrier Engrg. Corp., Cleveland, Ohio.	H. B. Matzen H. A. Thornburg	L. G. Powers L. A. Walton
FEYDT, JOHN CHARLES, Student, Carnegie Institute of Technology, Pittsburgh, Pa.	F. C. Houghten J. L. Blackshaw	C. M. Humphreys R. B. Stanger
FILKINS, HARRY L., Vice-Pres., City Ice Co. of Kansas City, Kansas City, Mo.	E. K. Campbell M. D. Woodling	F. F. Dodds D. D. Zink
FITZSIMONS, JAMES PATRICK, Student, Carnegie Institute of Technology, Pittsburgh, Pa.	J. L. Blackshaw F. C. Houghten	C. M. Humphreys R. B. Stanger
GILFRIN, GEORGE FREDERICK, Westinghouse Electric International, Mexico City, Mexico	A. E. Stacey, Jr. J. M. Rachal	H. L. Janet Herman Worsham
HENDRICKSON, JOHN JOSEPH, Asst. Chief Engr., Bryant Heater & Mfg. Co., Cleveland, Ohio.	W. E. Stark J. E. Kinner	Otto Fox J. N. Schweikert
HERRMANN, HAROLD C., Research Fellow, University of Minnesota, Minneapolis, Minn.	F. B. Rowley A. B. Algren	M. S. Wunderlich Albert Buenger
HICKEY, JAMES WILLIAM, Student, Carnegie Institute of Technology, Pittsburgh, Pa.	C. M. Humphreys F. C. Houghten	J. L. Blackshaw R. B. Stanger
JACKSON, ALTON B., New England Dist. Mgr., Carrier-York Corp., Winchester, Mass.	David Moulton Leslie Clough	J. W. Brinton G. B. Gerrish
JOHNSTON, JOHN F., JR., Johnston Bros., Inc., Ferrysburg, Mich.	C. H. Alexander W. W. Bradfield	O. D. Marshall P. J. Busch
MACMAHON, WILLIAM KENNEDY, Asst. Mgr. Utilization Dept., Washington Gas Light Co., Washington, D. C.	Leon Ourusoff E. F. Miller *A.S.M.E.*	David Allen *A.S.M.E.* W. M. Russell *A.I.C.E.*
MELICHAR, WOLFGANG J., 3959—44th St., Long Island City, N. Y.	T. W. Reynolds R. E. Daly	W. R. Zuhlke H. G. Ullman
MAXWELL, GEORGE W., Student, Carnegie Institute of Technology, Pittsburgh, Pa.	F. C. Houghten J. L. Blackshaw	C. M. Humphreys R. B. Stanger
PESTERFIELD, CHARLES HENRY, Research Fellow, Heating & Ventilating, University of Minnesota, Minneapolis, Minn.	F. B. Rowley A. B. Algren	M. S. Wunderlich Albert Buenger
PFEIFER, OTTO JOHN, JR., Insulite Co., Minneapolis, Minn.	F. B. Rowley A. B. Algren	M. S. Wunderlich Albert Buenger
PRICE, ERNEST HARRY, Student, Carnegie Institute of Technology, Pittsburgh, Pa.	F. C. Houghten J. L. Blackshaw	C. M. Humphreys R. B. Stanger
SCHLICHTING, WALTER G., Mgr., Air Conditioning Dept., Clarage Fan Co., Kalamazoo, Mich.	C. R. McConner S. I. Rottmayer	Albert Pelletier S. R. Lewis
SCHWERTFEGER, ANTON, Research Fellow, University of Minnesota, Minneapolis, Minn.	F. B. Rowley A. B. Algren	M. S. Wunderlich Albert Buenger
SMALL, BARTLETT RAY, Office Engr., Carrier-York Corp., Charlotte, N. C.	H. P. Gant J. A. Carey	C. D. Graham C. E. Scott
SWANSTRON, ALFRED E., 1444 Van Buren St., St. Paul, Minn.	F. B. Rowley A. B. Algren	M. S. Wunderlich Albert Buenger
VINSON, NEAL LORING, Student, Carnegie Institute of Technology, Pittsburgh, Pa.	F. C. Houghten J. L. Blackshaw	C. M. Humphreys R. B. Stanger

Candidates Elected

In past issues of the JOURNAL of the Society the names of the following men were listed as Candidates for Membership. The membership grade of each Candidate has been assigned by the Membership Committee and balloted upon by the Council. We are now instructed by the Council to post herewith, as required by Art. II, Sec. VIII, of the By-Laws, the following list of candidates elected:

MEMBERS

ELLIS, FREDERICK E., Imperial Iron Corp., Ltd., Toronto, Ont., Canada. (*Reinstatement*)

DUNCAN, JAMES R., Sales Engr., Carrier-Australasia, Ltd., Sydney, Australia. (*Reinstatement*)

FOX, OTTO, Vice-Pres., Bryant Heater & Mfg. Co., Cleveland, Ohio.

HUMPHREYS, CLARK M., Carnegie Institute of Technology, Pittsburgh, Pa.

RYDELL, CARL A., Carrier-Lyle Corp., Buffalo, N. Y. (*Advancement*)

SCHWEIKERT, JOHN N., Vice-Pres., Bryant Heater & Mfg. Co., Cleveland, Ohio.

SLAYTER, GAMES, President, Games Slayter, Inc., Cons. Engrs., Detroit, Mich.

STEPHENSON, L. A., Mgr., Powers Regulator Co., Kansas City, Mo. (*Reinstatement*)

TARR, HAROLD M., 21 Montgomery St., Arlington Heights, Mass.

TEMPLE, WALTER J., J. A. Temple Co., Kalamazoo, Mich.

Heating · Piping and Air Conditioning

EDITORIALS

Three Important Events

Heating, air conditioning, ventilating and refrigerating engineers will find much of interest and value in Cleveland the latter part of this month, when the annual meetings of the *American Society of Heating and Ventilating Engineers* and the *American Society of Refrigerating Engineers,* and the *International Heating and Ventilating Exposition,* will be held. The programs of both societies seem to be of particularly practical value this year, and the papers which will be presented are very timely. The exposition will contain an imposing group of exhibits.

Probably at no time in many decades has competition for every one been as keen as it is these months. Purchases are scrutinized more closely and consequently the engineer finds it necessary, even more than usually, to design, install and operate mechanical equipment which will provide the needed service at the minimum cost. This requires thorough knowledge of the latest practices and familiarity with improved types of apparatus which are developing all the time.

Attendance at such meetings and expositions as these apparently will be should prove of considerable value to the engineers in these fields.

Piping Economies

It is usually possible to judge whether or not a plant is being operated efficiently by an inspection of the piping. First of all, the question of flexibility is being given so much more attention these days by designers, the technical press, etc., that any plant operator should be conscious of the fact that piping must be designed and installed for proper flexibility or expensive maintenance will result. In more than one plant tight joints will never be possible because of faulty design to take care of expansion and contraction; hence, a never-ending expense is present in steam loss through leaks.

Another important point to remember is that the best expansion joints will not function unless repacked at regular intervals and lubricated as often as operating conditions require. It is of further interest to note that many lines have expansion joints or expansion bends, but no provision for proper alignment of the piping. Some engineers simply insert the proper number of joints or bends and forget entirely about the necessary anchoring and guiding of the lines. Without this anchoring and guiding, the expansion joints or bends often prove worthless as far as taking care of expansion and contraction is concerned.

Many plants are losing steam constantly through leaking stuffing boxes on valves. Such leaks not only provide a source of loss of steam, but in time often ruin valuable covering, etc. In a properly operated plant there is no excuse for valves leaking around the stuffing boxes.

Broken insulation caused by an employe stepping on pipe covering to operate a valve represents bad practice, because it is a simple matter to install chain wheels on valves in inaccessible places. The small cost of chain wheels or geared operation more than offsets such danger. An added advantage in some cases is secured because, with chain wheels, a valve can be operated quickly.

The improvement of industrial piping systems offers a fertile field for the reduction of maintenance and operating costs. In many plants, a careful inspection and study would quickly reveal numerous opportunities where economies can be made.

Air Conditioning Essential

No air conditioning system was provided in a large office building erected not many months ago. Last summer, it was found essential to provide conditioning equipment for several areas, including a large restaurant on the first floor. This was done, but required considerable ingenuity to find available space for the compressors, evaporator, etc.

According to the management of the restaurant, it would have been impossible to operate successfully during the summer months had not the air been cooled and dehumidified.

There are few buildings of this type to which air conditioning can not bring advantages. Especially for those enterprises that must depend on attracting the public every day, the provision of comfortable atmospheric conditions seems essential. By making provisions for air conditioning in the original plans, the cost of the installation is lowered.

A Wasteful Practice

Quite often when the vacuum is low on an extensive system of heating return lines, the excuse is, "The vacuum pump is no good," when it is only necessary to get busy and locate leaks in the return pipe lines or steam traps, obstructions in return pipe connections, drip valves open to the atmosphere, and other faults.

A few traps blowing through will fill the return lines with steam and the heat from this steam will cause other thermostatic traps to close and interfere with the heating at other points. Frequently in a heating system consisting of several buildings, when one building fails to heat properly, the trouble may be in the equipment of an adjacent building. Just because a radiator is hot is not always a true sign that it is heating efficiently. It may have a steam trap leaking that will cause trouble to other units; it is also a wasteful and expensive practice.

The Monday Morning Mail

As this editorial is being written, desks are laden with letters from subscribers; letters asking questions, letters discussing engineering problems, letters commending articles, letters questioning or criticizing articles, letters offering side-lights on current or unusual engineering problems, letters suggesting improvements in methods of calculation or procedure. Some are intriguing, some inspiring, some filled with curiosity, some informative and some prophetic.

Among these letters is one which says merely: "We want to take advantage of this holiday season to thank you for providing us with such an excellent journal."

We, on the other hand, wish to express our gratitude to the writers of these many letters for the interest manifested and for the help and inspiration they contain. For it is not only a pleasure to receive such letters,—they are of great value and a most important factor in the providing of useful technical material. In fact, an excellent measuring stick of the value of the service rendered by a publication is found in the letters received from its readers. The benefits are mutual.

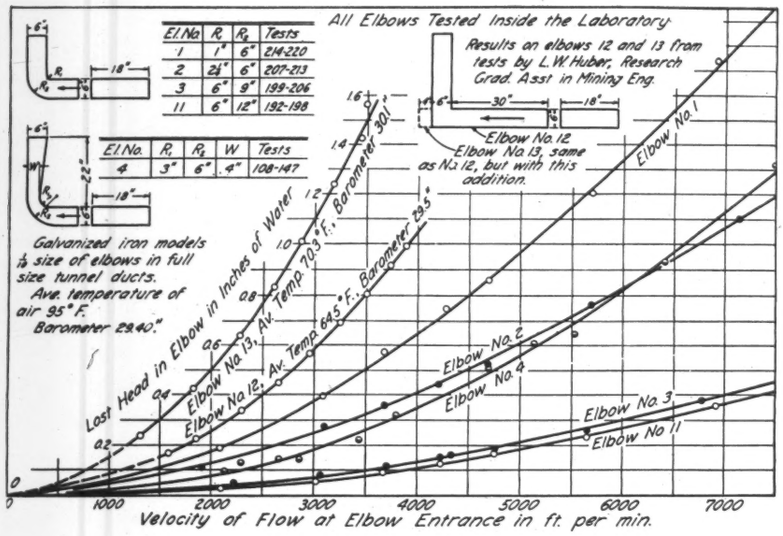

All Elbows Tested Inside the Laboratory
Results on elbows 12 and 13 from tests by L.W. Huber, Research Grad. Asst in Mining Eng.
Elbow No. 12
Elbow No. 13, same as No. 12, but with this addition.
Galvanized iron models ½ size of elbows in full size tunnel ducts. Ave. temperature of air 95° F. Barometer 29.40".
Elbow No. 13, Av. Temp. 70.3° F., Barometer 30.1
Elbow No. 12, Av. Temp. 64.5° F., Barometer 29.5
Elbow No. 1
Elbow No. 2
Elbow No. 4
Elbow No. 3
Elbow No. 11
Lost Head in Elbow in Inches of Water
Velocity of Flow at Elbow Entrance in ft. per min.

FIG. 1

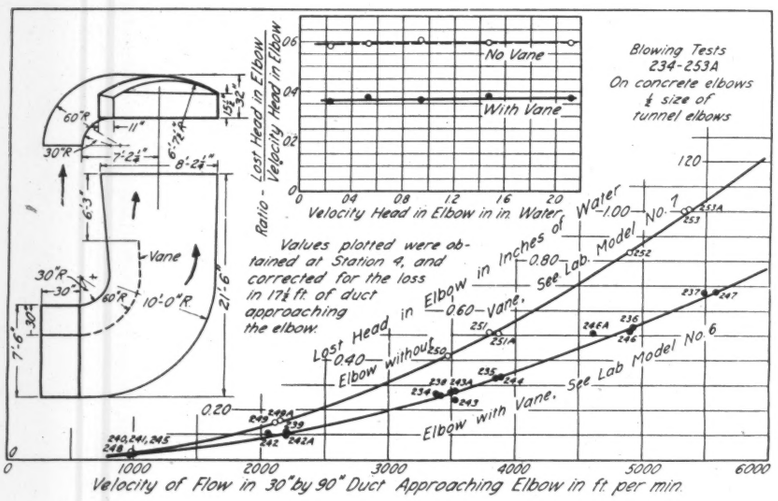

No Vane
With Vane
Blowing Tests 234-253A On concrete elbows ½ size of tunnel elbows
Ratio - Lost Head in Elbow / Velocity Head in Elbow
Velocity Head in Elbow in in. Water
Values plotted were obtained at Station 4, and corrected for the loss in 17½ ft. of duct approaching the elbow.
Vane
Elbow without Vane.
Elbow with Vane.
See Lab Model No. 6
See Lab. Model No. 7
Lost Head in Elbow in Inches of Water
Velocity of Flow in 30" by 90" Duct Approaching Elbow in ft. per min.

FIG. 2

"Open for Discussion"

IN ORDER to obtain either additional information or interesting discussion, requests for comment on F. E. Wertheim's article, "Turning Blades Reduce Duct Elbow Losses" (see November, 1931, HEATING, PIPING AND AIR CONDITIONING) were sent to several engineers. Among the replies are the following:

Comment from S. R. Lewis[1]

"The original study which developed the truth concerning elbow losses and the value of interior vanes was made, I believe, by Loring Wirt in connection with the steam flow to and from low pressure turbines.

"The information in Mr. Wertheim's article has been in the hands of engineers for some time. We have designed all our work in this office in accordance with the Loring Wirt data for several years."

[1] Consulting engineer, Chicago, Ill.

Comment from A. C. Willard[2]

"I read this article with interest but have no particular comments to make concerning the matter presented.

"Some time ago we prepared a brief article on friction losses in elbows for the college of engineering publication *The Technograph* and I am enclosing a reprint of that article for your information."

Closure by F. E. Wertheim[3]

It does not take very extensive investigation to discover that there are far too many air duct systems and fan or blower installations which could be materially improved by reconstructing duct elbows. This fact is well illustrated by reference to Professor Willard's

[2] Professor, Heating & Ventilation and Head of Department of Mechanical Engineering, University of Illinois, Urbana, Ill.
[3] Member, *Western Society of Engineers.*

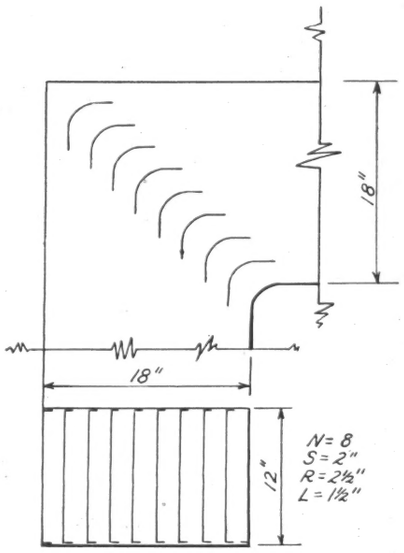


FIG. 3—ELBOW WITH TURNING BLADES (BASED ON LORING WIRT FORMULA)

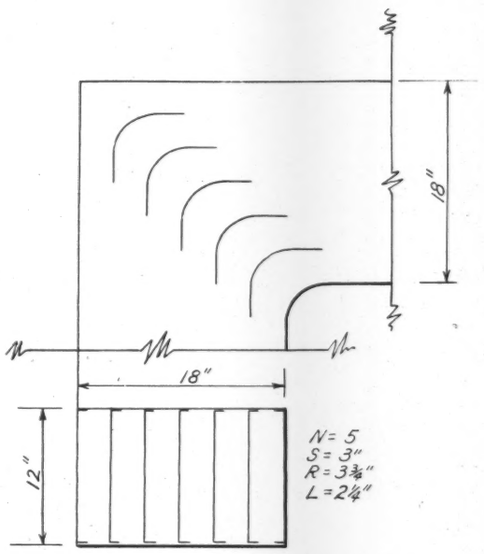


FIG. 4—ELBOW WITH TURNING BLADES (BASED ON B. L. SPIVAK FORMULA)

article in *The Technograph* in which the cost of uneconomical operations due to a type of elbow commonly used at the base of down cast shafts in mines is figured, showing a possible saving of $4,300 per annum. This elbow is No. 13 on the chart, Fig. 1, which was published in *The Technograph* and also appears in "Mechanical Equipment of Buildings," Vol. 1, by Harding and Willard, page 674.

The writer wishes to remark, however, that while the elbow of two-piece construction, i. e., with both inside and outside made square (No. 12) has a very poor performance curve, the experiments of Mr. Wirt mentioned by Mr. Lewis show that best results were obtained with an elbow having a round bend at the inner side, or throat, and a sharp mitered corner for the outer bend.

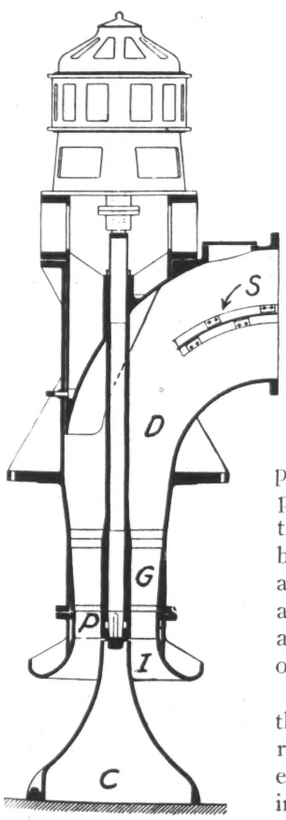


FIG. 5—VERTICAL PROPELLER TYPE PUMP WHICH ILLUSTRATES THE IMPORTANCE OF GUIDING WATER OR AIR FLOW *C*, TUNNEL GUIDE CONE; *I*, BELL-SHAPED INLET; *P*, PROPELLER; *G*, GUIDE VANES; *D*, DIFFUSER PUMP BODY; *S*, SPLITTER IN DISCHARGE ELBOW

The Technograph article further shows the value of large radius ratios, of venturi-style elbows, and of a splitter used in a compound elbow as shown in Fig. 2. Prof. Willard's article is an abstract of his investigations and research undertaken in connection with the ventilation of the Holland vehicular tunnel.

The chart given on page 917 of the November, 1931, HEATING, PIPING AND AIR CONDITIONING varies somewhat from the proportions recommended by Mr. Wirt. Figs. 3 and 4, drawn to scale, show 18-in. x 12-in. elbows. Fig. 3 shows the elbow designed according to Mr. Wirt's data. The design shown in Fig. 4 and based on the information given in my article might be equally good.

Recently, elbows with turning blades have been used in some outstanding power plants, such as the James H. Reed Station of the Duquesne Light Co., near Pittsburgh, in which square corners, turning vanes, and venturi ducts are incorporated in the breechings. In this same station are some low-head condenser circulating pumps, Fig. 5, which are mentioned here because they show so clearly the analogy in the flow of air and water. The good provisions for flow throughout this pump are noteworthy, as witness the splitter in the discharge elbow.

The writer wishes to thank Messrs. Willard and Lewis for their comments. The subject of pressure drop in elbows has long been investigated; nevertheless the phenomena of flow in elbows of different radii are still the subject of careful consideration. For those interested in details, I would mention a very interesting discussion in paper F S P—53—17, *Trans. A. S. M. E.*, 1931.

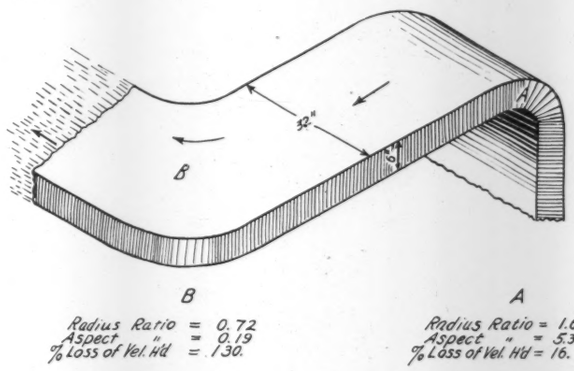


FIG. 6—THIS SKETCH TELLS THE STORY. *A* IS AN EFFICIENT ELBOW; *B* IS INEFFICIENT. SQUARE OUTER CORNER AND TURNING BLADES WOULD IMPROVE *B* MARKEDLY

Paints Indicate Approximate Temperature

EXPERIMENTAL or routine work that requires the approximate measurement of temperatures of surfaces or of parts often can be facilitated by the use of paints that change color with temperature. A part of a machine, such as a bearing, gear, or housing, or piping, tanks, ducts, etc., if covered or striped with such a paint will indicate its approximate temperature.

Composition of a Paint for Temperatures to 300 F

There are several paints of this character, one of the best known using the double iodide of copper and mercury as a pigment and a light colored non-acid varnish as the vehicle. One part of cuprous iodide, by weight, is mixed intimately with two parts of the mercuric iodide, either dry or wet. If wet mixed, the water must be evaporated by gentle heating before the pigment is ground and mixed in the vehicle. A paint made in this way is bright red at room temperatures and up to 130 F, at which a perceptible darkening occurs. Another rise of five degrees results in noticeable darkening, and at 145 F the color is maroon. A further rise of ten degrees brings out a light chocolate and at 160 F the color is a dark chocolate. At 190 F the color becomes the darkest that can be distinguished and at the boiling point of water the paint is black. Upon cooling, the colors reappear in the reverse order but at the same temperatures, except that between 212 and 170 F the colors are difficult to determine. This cycle, up or down, can be repeated indefinitely, so long as the paint is not heated above 300 F, at which point decomposition begins.

This temperature scale was worked out by S. W. Rushmore and the same or a similar paint has been made and used in the laboratory of John W. Masury & Son, paint manufacturers.

Mr. Rushmore has used this paint on pipes, radiators and parts of engines involved in steam heating and cooling systems with which he has experimented and has found it most useful for this purpose. When applied to a radiator, for example, the paint shows which portions are in contact with live steam, as these will be black, and which contain cooler water or air, the temperatures of which are also indicated by the color. It was found inadvisable to coat an entire radiator in a closed room, however, as the paint gives off an unpleasant odor when heated. A narrow stripe or spots serve the same purpose.

In heating, power plant, or air conditioning work, especially where some trouble occurs or where some experimenting may be required to correct difficulties encountered, a pipe, tank or duct may be painted with a stripe of the paint and the temperature will be indicated by the color of the stripe, providing the temperatures come within the range named. Thus, a part which turned the paint black would be known to be above 190 F and one which remained red would be at 130 F or below, perhaps indicating where an air-bound section existed.

In the case of a water heater tank, a stripe of the paint will indicate the various levels at which the water is hot or cool, and if used on the pipes of a heating system will show at a glance which pipes are cold, which are above 130 F and which are at higher temperatures. Ventilating ducts for hot air, unit heaters, and even stacks or chimneys that do not attain a temperature above that at which the paint decomposes can be painted to indicate the temperature. Other applications might include pipes, tanks, and other equipment in factories where steam or other heating at moderate temperatures is required. Motors or bearings, which may have a tendency to overheat through overloads or otherwise, may have a spot or stripe of the paint applied as a danger signal.

Another Temperature-Sensitive Paint

A similar temperature-sensitive paint can be made from the double iodide of silver and mercury, which is yellow when cold but turns to orange or brick red when heated to a temperature of 160 to 175 F. In this case the proportions recommended are five parts of silver iodide to one part of mercuric iodide.

Several other salts, including some containing mercury, copper, arsenic, lead, tin or iron, change color when heated, but the two double iodides mentioned are probably the best-known compounds of this kind.

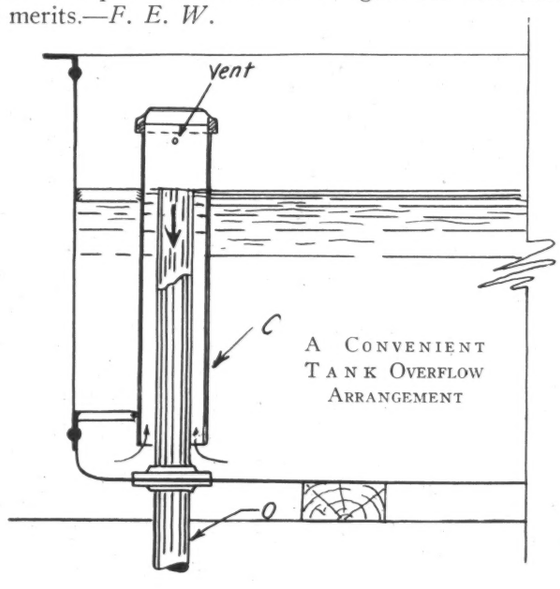
Herbert Chase.

A Simple and Practical Tank Overflow

PERFORATED strainers, as a means of preventing the escape of sediment from tank outlets, are often most troublesome and unsatisfactory because of the difficulty of keeping them clear of deposits tending to choke the outlet.

In the sketch, the tank is fitted with an overflow pipe *O* passing through the tank bottom. A pipe *C* surrounds the overflow pipe and prevents any surface scum or particles from entering the overflow pipe. All sediment heavier than the liquid will, of course, settle at the bottom. The vent hole is essential, otherwise the tube will act as a siphon; but if desired the end of pipe *C* may be left open instead of being provided with cap and vent hole.

The arrangement shown gives full area of overflow pipe while functioning satisfactorily as a strainer. Moreover, the construction is simple, cheap and efficient. This is an old device which seems to have been forgotten—hence it is presented in order to again call attention to its merits.—*F. E. W.*



A Convenient Tank Overflow Arrangement

Practical Piping Problems

Properly Located Shut-off Valves Aid Maintenance

W. H. Wilson's Monthly Discussion of the Problems He Meets From Day-to-Day

MANY otherwise well-designed piping installations are lacking in shut-off valves at the proper locations to close off sections of the pipe lines if necessary when repair work is done. This is due to neglect in providing a sufficient number of sectional cut-off valves at the time the piping is installed or to valves not being located at the proper places. Alterations and changes to a piping system may require the installation of additional sectional cut-off valves.

Location of Valves Depends on Conditions

Selecting the number and location of these valves depends on local conditions and the kind of service that is required of the pipe lines. For underground water lines the nature of the soil should be considered. Water mains laid in hard clay soil are free from many of the causes of pipe lines breaking and leaking that are present when pipes are laid in loose, filled-in ground, or in lines that pass under railroad tracks, through building foundation walls, in cinder fill, etc.

Where there is a possibility of underground mains being put out of service on account of such conditions, sectional cut-off valves should be installed in these areas. For instance, a pipe line that passes under the bed of a river or stream should be provided with a shut-off valve on each side of the stream.

Fire Protection Piping Changed

Fig. 1 represents the location of underground water supply mains to automatic fire protection sprinklers for a large three-story building of wood construction.

A truck struck the post indicator for operating the underground valve (shown at *M*) controlling the supply riser for that section of the building. This accident caused the bonnet of the valve to be broken and bent the stem, requiring a new valve to be installed. After the ground was removed and the valve exposed, it was found to be a spigot-end valve placed between two elbows and located near concrete fillings of former foundations. The installation of the new valve was not a simple matter.

By extending the supply a short distance, as shown by the dotted lines, water is more evenly distributed and

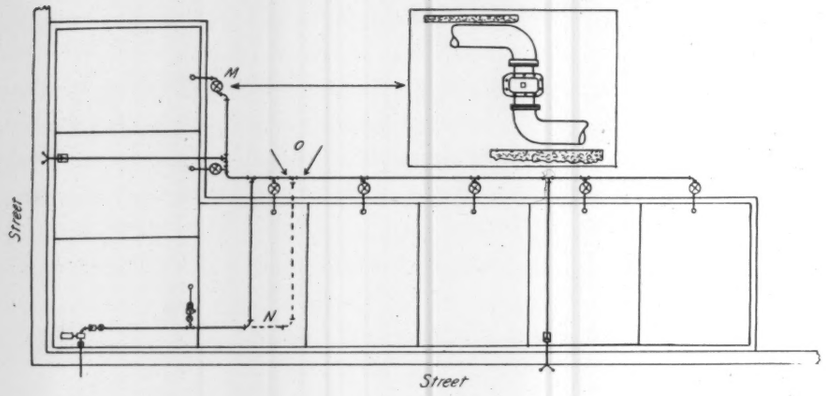


FIG. 1—BY CHANGING THE SUPPLY LINE AS SHOWN BY THE DOTTED LINES AND INSTALLING SHUT-OFF VALVES AT *O*, WATER PRESSURE CAN BE KEPT ON ONE SECTION OF THIS FIRE PROTECTION PIPING WHEN REPAIRS ARE MADE ON THE OTHER

the installation of shut-off valves at the two locations, shown at *O*, permits water pressure to be kept on part of the building when repairs to sections of the supply main must be made.

Fig. 2 is a general plan of the underground fire protection water mains for a large plant consisting of a number of buildings.

At the time the water lines were installed the designer insisted on loop or circuit mains, with a sufficient number of sectional cut-off valves in order that the fire protection service should not be impaired to any great extent in case of repairs to the underground piping or hydrants. This is good practice. The pump house, however, was located some distance from the main buildings and the entire water supply from the pumps depended on the one pipe line shown at *P*, which was laid in soil not favorable for the location of underground lines. Fig. 3 shows a re-arrangement of this lay-out, with the pump discharge for the water supply arranged so that two lines are provided. In case one should fail the arrangement of valves will permit this particular line to be closed off and the water supply from the pumps will not be completely impaired.

The fire department pumping connection *S* is also relocated. It can be ready for use in case different sections of the pipe lines are closed off.

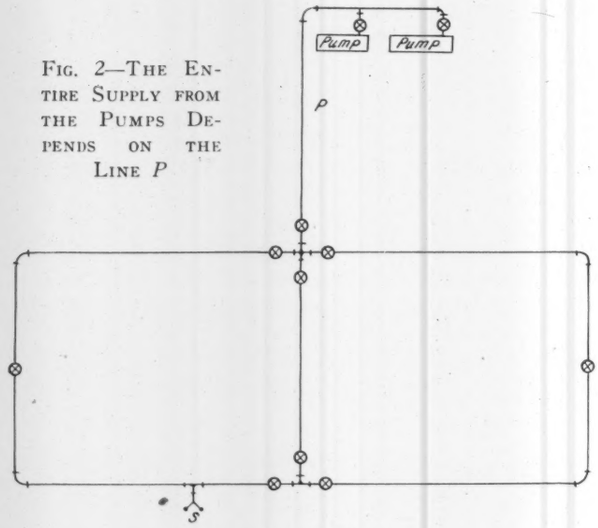


FIG. 2—THE ENTIRE SUPPLY FROM THE PUMPS DEPENDS ON THE LINE *P*

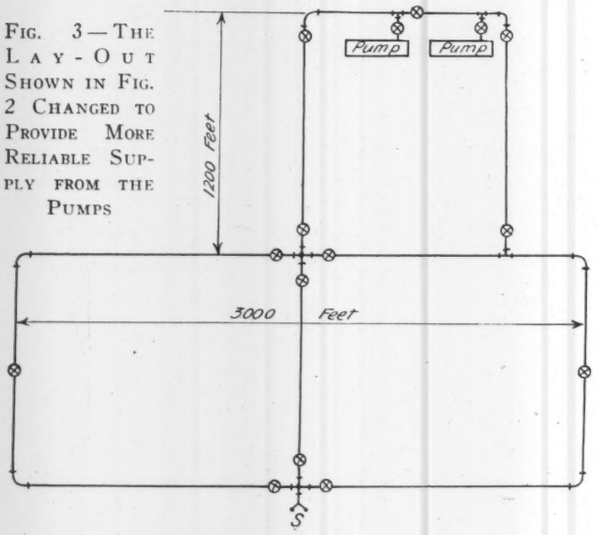


FIG. 3—THE LAY-OUT SHOWN IN FIG. 2 CHANGED TO PROVIDE MORE RELIABLE SUPPLY FROM THE PUMPS

Shut-off Valve in Steam Header

Fig. 4 represents the main steam header for a boiler house, the piping being a loop provided with shut-off valves. A division valve is shown at *G*, so that either group of boilers can be kept in operation, in case it is necessary to close off part of the header.

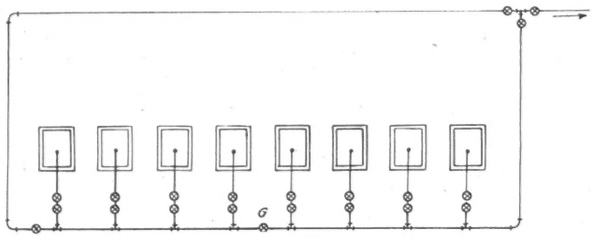


FIG. 4—DIAGRAM OF MAIN STEAM HEADER IN A BOILER PLANT, SHOWING DIVISION VALVE AT *G*

Re-arrangement of Pump Suction Line

This boiler house was also equipped with two independent steam supply lines to operate the boiler feed pumps and two independent feed water lines to supply the boilers. Only one suction line was provided for the boiler feed water pumps from the feed water heater.

FIG. 5—SUCTION LINE FOR BOILER FEED WATER PUMPS

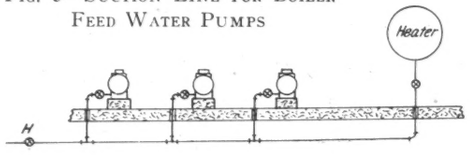


Fig. 5 shows the arrangement of this suction line, with the emergency, or cold water supply, shown at *H*. This suction is a straight pipe line with the connections to each pump passing directly through a concrete floor, making a rigid pipe line. In case it is necessary to repair this pipe line or renew a gasket, very little movement can be made in order to clean the faces of the flanges and insert a new gasket.

As the rest of the piping is arranged so that sections of it can be closed off without interfering seriously with the operation of the boilers, this suction line should be arranged in a similar manner.

Fig. 6 shows a different method of arranging the suction piping, the ends of the pipes being at *K* where the emergency water connection can be provided for each line. A twin supply to each pump is provided and the piping arranged so as to be more convenient for repair work on either header.

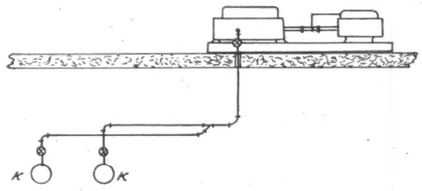


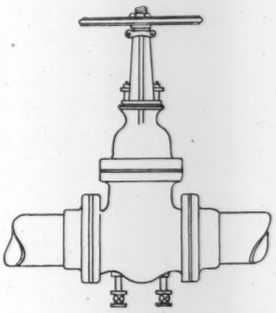
FIG. 6—THE ARRANGEMENT SHOWN IN FIG. 5 CHANGED SO THAT REPAIR WORK MAY BE DONE ON EITHER LINE

Drain Tappings on Gate Valves

Fig. 7 shows a valve for a steam line with a drain tapping on either side of the gate. When turning on a section of a steam header it is safe practice to open a drain at the opposite end to relieve the water that will accumulate on account of a certain amount of condensation from the pipe line.

When the water flows out and the steam pressure is equalized, both of the large valves can then be safely opened to their full extent. For large steam headers that are continually in use these drain valves are a great help. In case it is necessary to close a division valve for any appreciable length of time, a steam trap can be temporarily installed on the pressure side to prevent water accumulating and causing trouble.

FIG. 7—DRAIN TAPPINGS ON GATE VALVE



Drain valves for this purpose should be of good quality material, well made and installed with extra strong pipe nipples.

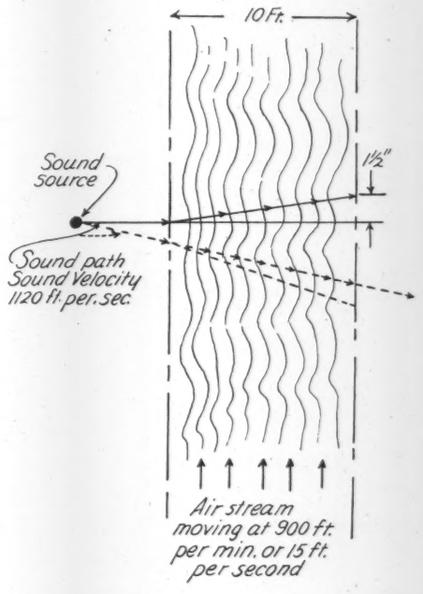
Air Velocity and Acoustics

REGARDING the effect of humidity and velocity in connection with acoustics, in laboratory work we find that the relative humidity of the air in the laboratory has a slight effect on the absorption coefficient of materials. In the investigation of the acoustical condition of an auditorium this feature is not usually considered.

There are many items entering into such an investigation whose value cannot be determined to a similar fine degree, hence it seems unnecessary to consider the humidity.

The velocity of ventilating air is very low compared to that of sound. Sound travels at 1,120 feet per second, ordinarily more than 75 times faster than ventilating air. Therefore the retarding effect of a current of air on sound in confined spaces is small. The illustration will clarify this statement. This difference in speed of sound and air also explains the need for lining ducts on *both* sides of a fan in a recirculating system.

One of the most annoying things in a theater or auditorium is the continued hum of a distant fan carried into the auditorium through the duct system. It is here that the ventilating engineer can be of real service by lining the ducts with noise absorbing materials. It is general practice to line the duct a distance of 10 to 15 times the least cross dimension with a material of about 50 per cent absorption, preferably at bends.—*E. F. Ries*, President, Acoustical Engineering Corporation, Chicago, Ill.



SINCE SOUND TRAVELS AT 1120 PER SECOND IT WOULD REQUIRE $\frac{10}{1120}$ SECONDS TO GO THROUGH THE AIR STREAM. IN THIS TIME THE AIR STREAM TRAVELING AT 15 FT. PER SEC. WOULD DISPLACE THE SOUND $\frac{10}{1120} \times 15 = 0.13$ FT., ABOUT 1½ IN.